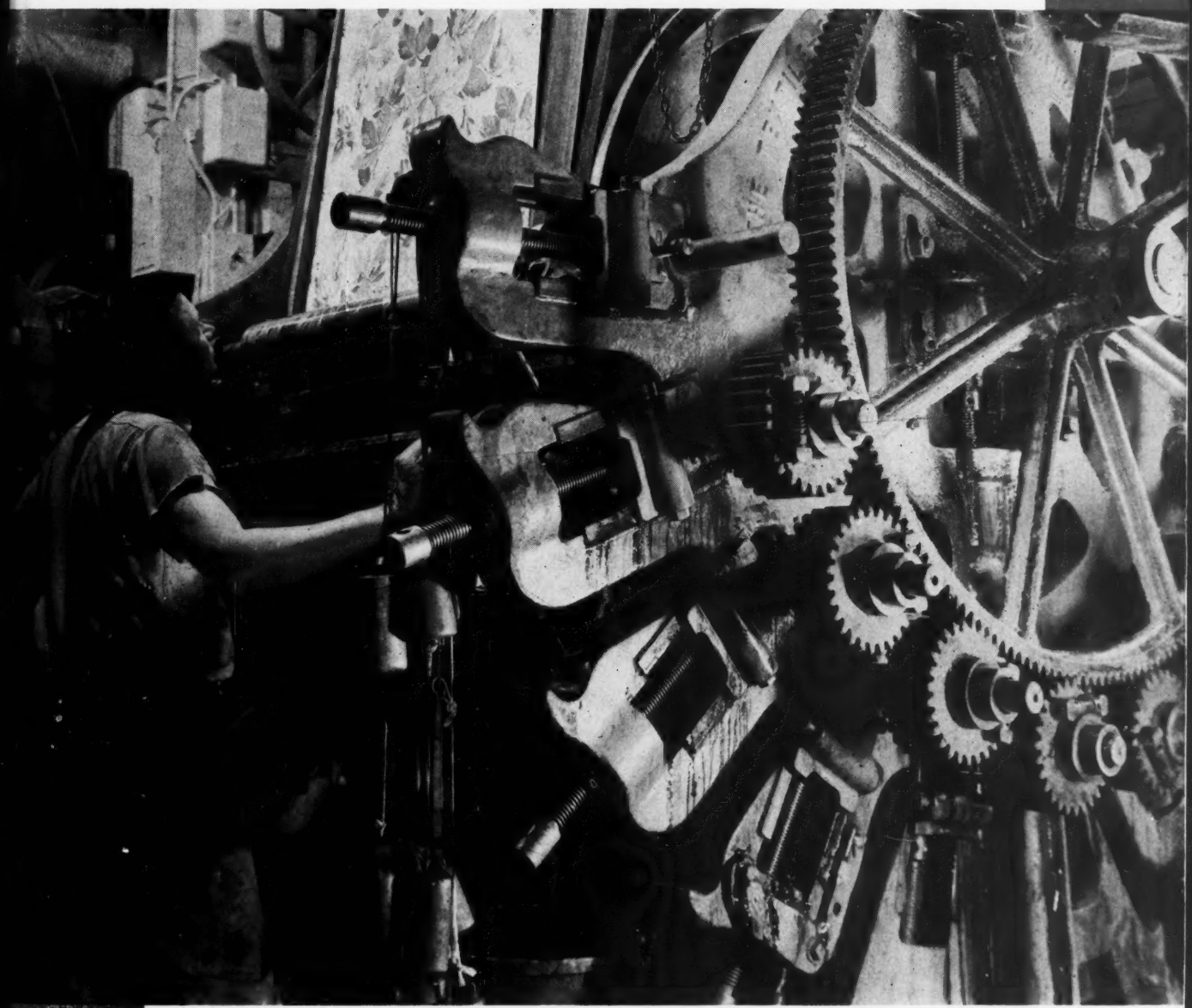
Vol. 13, No. 8 August, 1957

LUBRICATION ENGINEERING

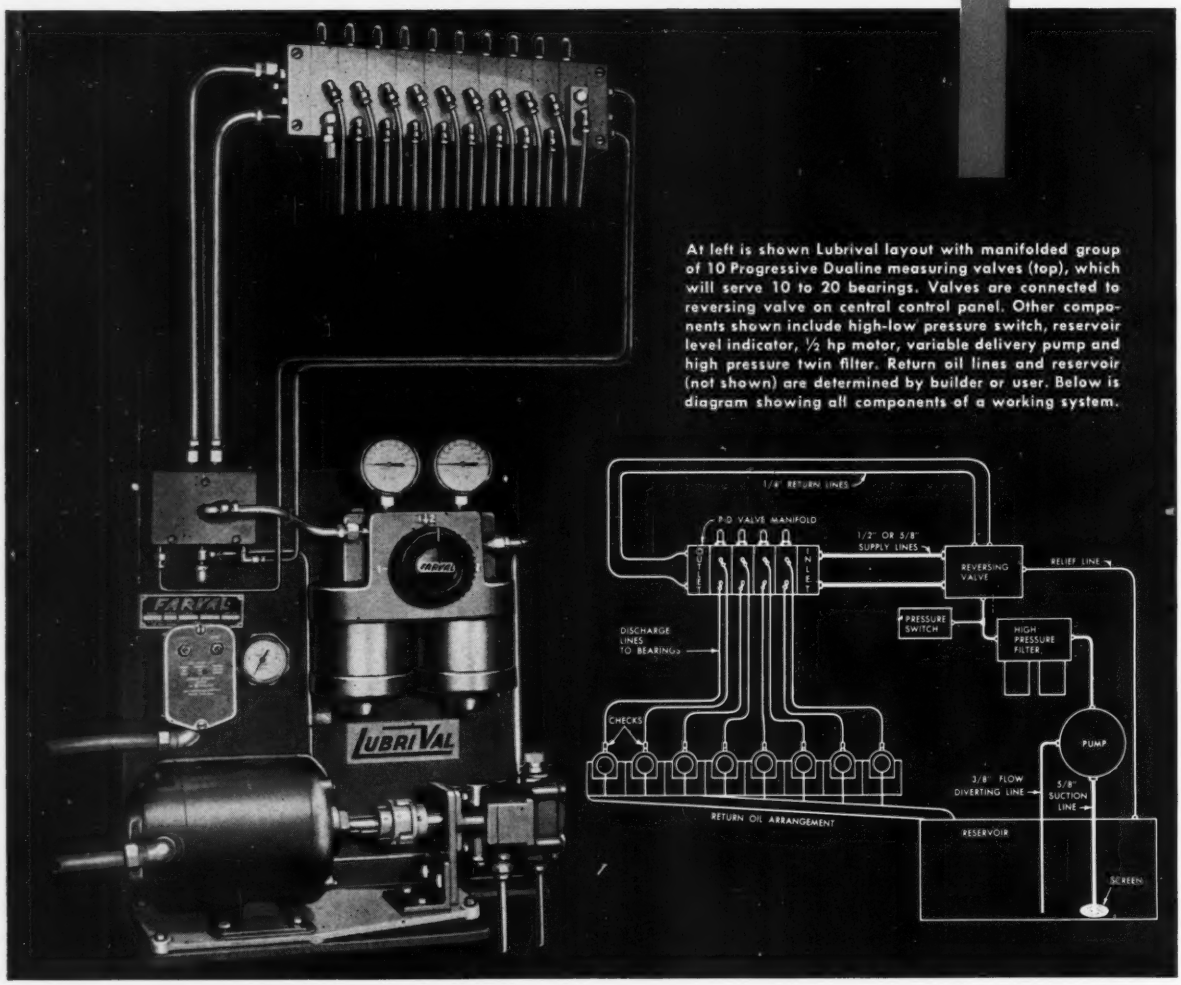
RESEARCH APPLICATION EDUCATION MAINTENANCE



JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

LUBRIVAL—a revolutionary new system to supply and circulate lubricating oil

FARVAL—
Studies in Centralized Lubrication
No. 204



At left is shown Lubrival layout with manifolded group of 10 Progressive Dualine measuring valves (top), which will serve 10 to 20 bearings. Valves are connected to reversing valve on central control panel. Other components shown include high-low pressure switch, reservoir level indicator, ½ hp motor, variable delivery pump and high pressure twin filter. Return oil lines and reservoir (not shown) are determined by builder or user. Below is diagram showing all components of a working system.

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LUBRICATION ENGINEERING

RESEARCH APPLICATION EDUCATION MAINTENANCE

Volume 13, No. 8

August, 1957

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Contents

APPLIED SCIENCE

Effective Lubrication at Chrysler 436
by D. F. Miller
Lubricating Oil Filtration on Diesel RR's 440
by W. King Simpson
Circulating Oil Systems: A Practical Manual 443
by R. E. Forsythe

RESEARCH, DESIGN, AND DEVELOPMENT

Synthetic Ester Lubricants 454
by M. Z. Fainman and R. S. Barnes
A Numerical Solution of Reynolds' Equation 459
by B. Sternlicht and H. J. Sneck
An Initial Axle Problem: Rear Axle Noise 464
by J. E. Cardillo

CURRENT NEWS

Lubrication in the News 424-28
News Notebook 468
Section News 470
Personals 473
Coming Events 476

FEATURES

The Sight Glass 422
Asperities 435
Lube Lines 447

LITERATURE SUMMARIZED

Current Literature 452-71
Summaries of Foreign Papers 450
Foreign Lubrication Abstracts 474
Compiled by Henry Brutcher
Patent Abstracts 442
Compiled by Ann Burchick

TRADE INFORMATION

New Products 430-32
Advertisers' Index 476

COVER PICTURE: Open Gear Lubrication in a Silk-Screen Textile Plant.
(Photo courtesy Gulf Petro-engineering Service)

Abstracted by Engineering Index

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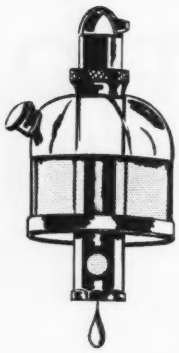
The Sight Glass



Highlights of Articles Scheduled for Coming Issues!

Cutting Fluid Evaluation by Accelerated Machining Tests

The performance of cutting fluids, both emulsifiable oils and compounded mineral oil products, has been difficult to evaluate. Often, selection of a product is on the basis of past performance of a similar nature. Using the radiometric tool wear and the tool force methods, cutting performance was investigated. Insert-type tools were used, and tool bits were irradiated with nuclear bombardment. Results of the tests show that tool wear for emulsifiable oils is greater than for the compounded mineral oils. No shift in the relative efficiency of the fluid was noted with increased speed.

Choose Your Press Drawing Lubricants with Care

Stamping and deep drawing operations are highly competitive, requiring that production costs be closely controlled. Problems not encountered in experimental runs develop during production runs and often these are attributed to inadequate lubrication. By understanding some of the basic mechanisms of the lubricant a rational approach to selection by lubricant type can be made.

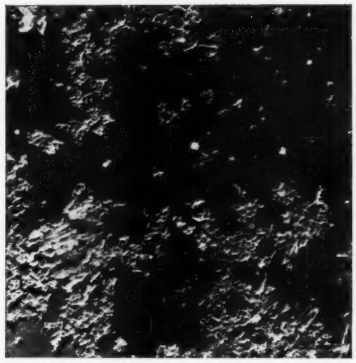
Fundamentals and Laboratory Evaluation of Cutting Fluids

Composition, hardness, heat treatment, and grain size of the work material; composition, hardness, and cutting angle of the cutting tool; cutting conditions, such as speed and depth of cut, and feed; these plus the role of the cutting fluid with respect to type, volume, pressure, and direction of flow are variables that affect the selection of the right fluid. Complete understanding of the cutting process and the nature of the lubrication requirements as described will place these variables in their proper relationship, resulting in a better basis for cutting fluid selection.

Performance of an Inorganic Microgel Grease in Industrial Applications

Minimum stocks of lubricants are possible whenever multipurpose lubricants, embodying properties which permit operation over a wide range of conditions and applications, can be used. One concept of multipurpose greases is based upon inorganic thickeners instead of conventional soap formulations. Field experiences for this type of product are detailed. Several superior applications are described.

LITHIUM Base Grease:



1μ

Micrograph showing soap structure of lithium base lubricating grease made from 12-hydroxy stearic acid. Courtesy of Sinclair Research Laboratories, Inc.

ITS MULTI-Properties MAKE IT MULTI-Purpose

IN CHARACTERISTICS

- Non-corrosive
- Water resistant
- High heat resistance
- Rheological properties
- Eliminates abnormal wear
- Soap oxidation stability
- Cold operating properties
- Oxidation and shear stability
- Mechanically and chemically stable

IN APPLICATION

- Prepacking anti-friction bearings
- Lubrication of plain and anti-friction rotating bearings
- Lubrication of vehicle chassis points, wheel bearings, universal joints and water pump
- Lubrication of high-temperature ball bearings
- Aircraft, artillery, instrument and general ordnance maintenance

IN MARKETING

- Lower inventory
- Less dispensing equipment
- No chance for misapplication
- Reduction in lubricating time
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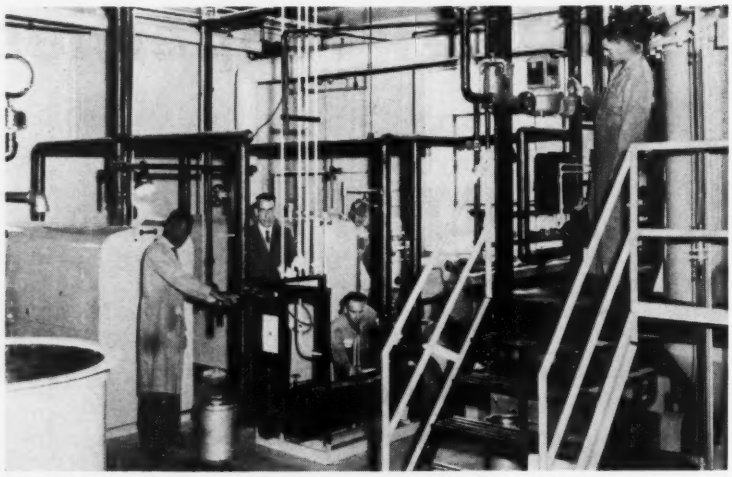
Automation Saves Manpower at First Canadian Lithium Grease Plant



Plant officials check blending unit at new Canadian lithium grease plant. Two "Votator" units are at left in background.

Labor saving through automatic instrumentation at critical process points has been achieved at the first continuous process for the production of lithium base greases in Canada. Placed on stream recently at the plant of Surpass Petrochemicals, Ltd., Toronto, the compact unit is intended to serve Canada's growing demand for lithium base greases for several years to come.

Due to efficient design, the unit has demonstrated production rates in excess of 3000 lb. per hour in spite of its small size. With strategic placement of automatic instrumentation, a single operator can run the unit with the part-time supervision of one chemical engineer.

Flexibility has been designed into the unit in two ways. First, the operating ranges of the variables are such that other greases can be produced in the same equipment, including both soap-base greases and organo-philic gel types. Second, small holdup and little back-mixing facilitate the rapid obtaining of steady state conditions, making short runs of special formula greases possible.

Close control of process variables during manufacture is emphasized, resulting in what plant engineers feel will be extremely stable structure in the greases being produced. In addition, short holdup time is said to minimize the degradation of oils and additives through oxidation, polymerization, cracking, and other undesirable reactions.

Gas Lubrication Research Begun at Franklin

PHILADELPHIA — Under a recent contract with the Office of Naval Research, the engineering staff of Franklin Institute Laboratories here has embarked on a program to "create a technology for gas-lubricated bearings." The three objectives of the program, as stated, are "to explain the fundamental phenomena in hydrostatic and hydrodynamic gas lubrication; to develop a sound design procedure for gas-lubricated bearings; to define the areas in which gas lubrication can be applied."

So far, continued the statement, research on gas bearings has singled out various effects at the cost of neglecting others; gas bearing design has commonly been a matter of cut and try. As a result, there is no general or inclusive theory for gas-lubricated bearings that can compare, for instance, with our understanding of plane bearings lubricated with incompressible lubricating fluids.

In a laboratory report issued by Franklin, it was predicted that gas bearings will become vitally important to designers of high-speed, high-temperature rotating machinery and machines for processing — without soiling or contaminating — food, chemicals, textiles, and drugs. The low-friction, temperature resistant and cleanliness properties of gas lubricants were cited as major advantages over liquid systems.

Texaco Plans Multi-Source Radiation Facility

What was described as one of the world's largest nuclear radiation laboratories is now under construction at the Texaco Research Center, Beacon, N. Y. Completion of the laboratory, which will eventually house three different radiation sources, is scheduled for April, 1958.

Nuclear test facilities of the completed project will include a 6 million volt linear accelerator, reputedly the first of its kind in the petroleum industry; a 3 million volt Van de Graaf generator, and a 35,000 curie Cobalt 60 gamma radiation source.

"Our program of nuclear studies is under expansion," stated Frederic H. Holmes, Texas Co. vice president in charge of research. In outlining the direction of future research, he added that "radioisotope investigations bearing on day-to-day problems in the use of lubricants and fuels will be increased. Our researchers hope to enlarge their understanding of how greases and oils lubricate, how wear and corrosion occur, why certain jet fuels are better than others, and other related problems."

Horizons To Develop WADC High-Temp Seals

A new contract from Wright Air Development Center, Dayton, Ohio, for the development of high temperature materials for rotating seals in jet aircraft engines has been awarded to Horizons Incorporated, Cleveland process and material research organization. Designs of Mach 2 and Mach 3 aircraft make it imperative that rotating parts such as bearings, gears and the like, for both engine and accessory components, be developed for 1000°F. temperatures.

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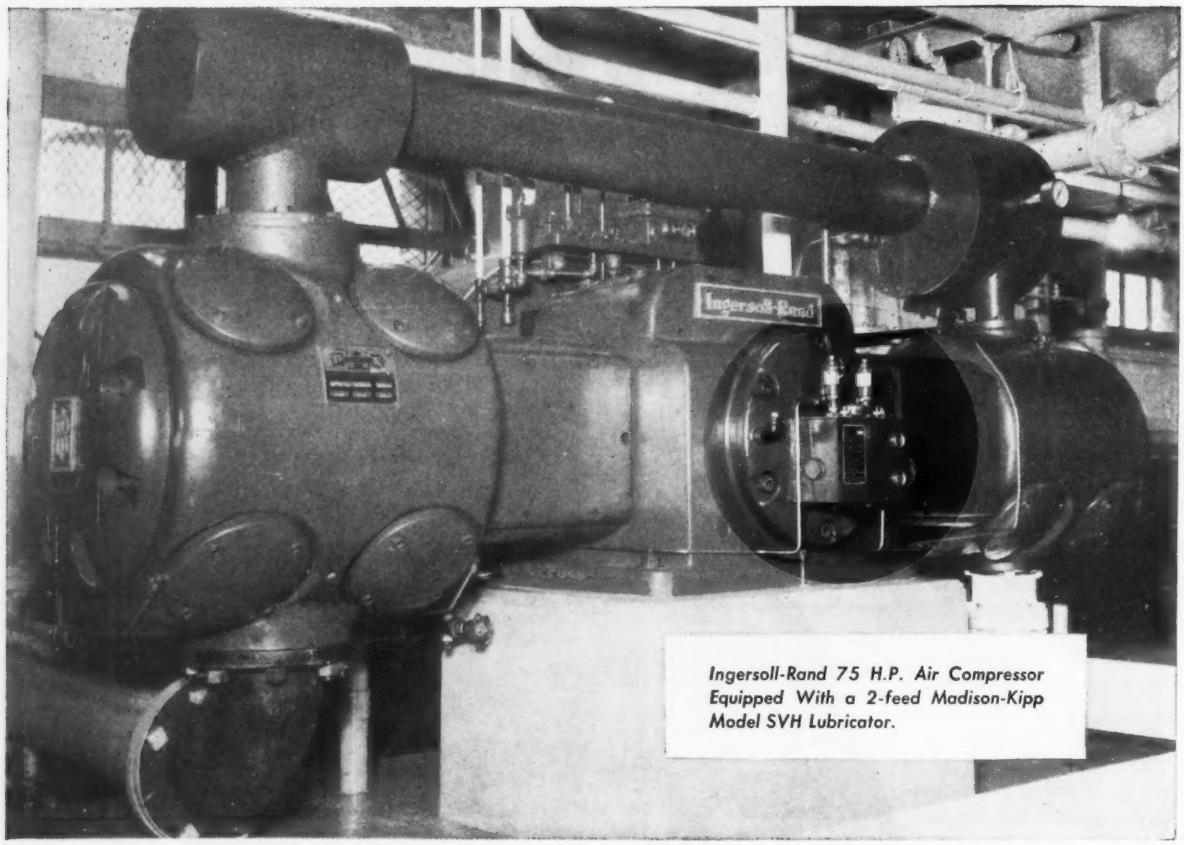

Ingersoll-Rand 75 H.P. Air Compressor Equipped With a 2-feed Madison-Kipp Model SVH Lubricator.

Machines of great performance use the most dependable oiling system ever developed

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New Army Fuel and Lubricant Research Lab Established in Southwest

SAN ANTONIO, TEX.—In line with increasing demand for research in military fuels and lubricants, a new Army Ordnance Fuels and Research Laboratory was officially dedicated here recently. Built on land leased from the Southwest Research Institute, the facility will be staffed and managed by personnel from that institution.

In a speech delivered at the dedication ceremonies, Major General August Schomburg, chief of Army Ordnance research and development, stressed the growing need for fuel and lubricant research, especially in military areas. He cited the presence of industry representatives as evidence of close industry-research team cooperation, and added that the laboratory will help define Ordnance needs, and then work with industry to meet these needs with products.

Among industry representatives present were Errol Gay, consultant; C. J. Livingston, Gulf Research and Development; Col. P. N. Gillon, Carborundum Company, and W. J. Sweeney, vice president, Esso Research and Engineering, Research and Development.

Research projects currently being conducted or planned by the laboratory include the following:

Survey of factors leading to vapor lock.

Utilization of JP-4 fuels in compression ignition engines — studies the effect of fuel characteristics on performance, starting, injection pump wear, spray patterns and deposits.

Studies of the role of oil soluble dispersants in maintaining colloidal suspensions and reducing engine deposits under light duty conditions.

Investigation of factors relating to oxidation stability and bearing corrosion characteristics of lubricating oils.

Correlation of engine manifold deposits with glass manifold results.

Thermal and oxidation stability of extreme pressure lubricant compositions.

Analysis of mechanism of extreme pressure lubrication for hypoid gears and development of improved mechanical evaluation apparatus.

ANALYTICAL LAB at SWI provides up-to-date instruments and equipment for Army Ordnance fuel and lube research.

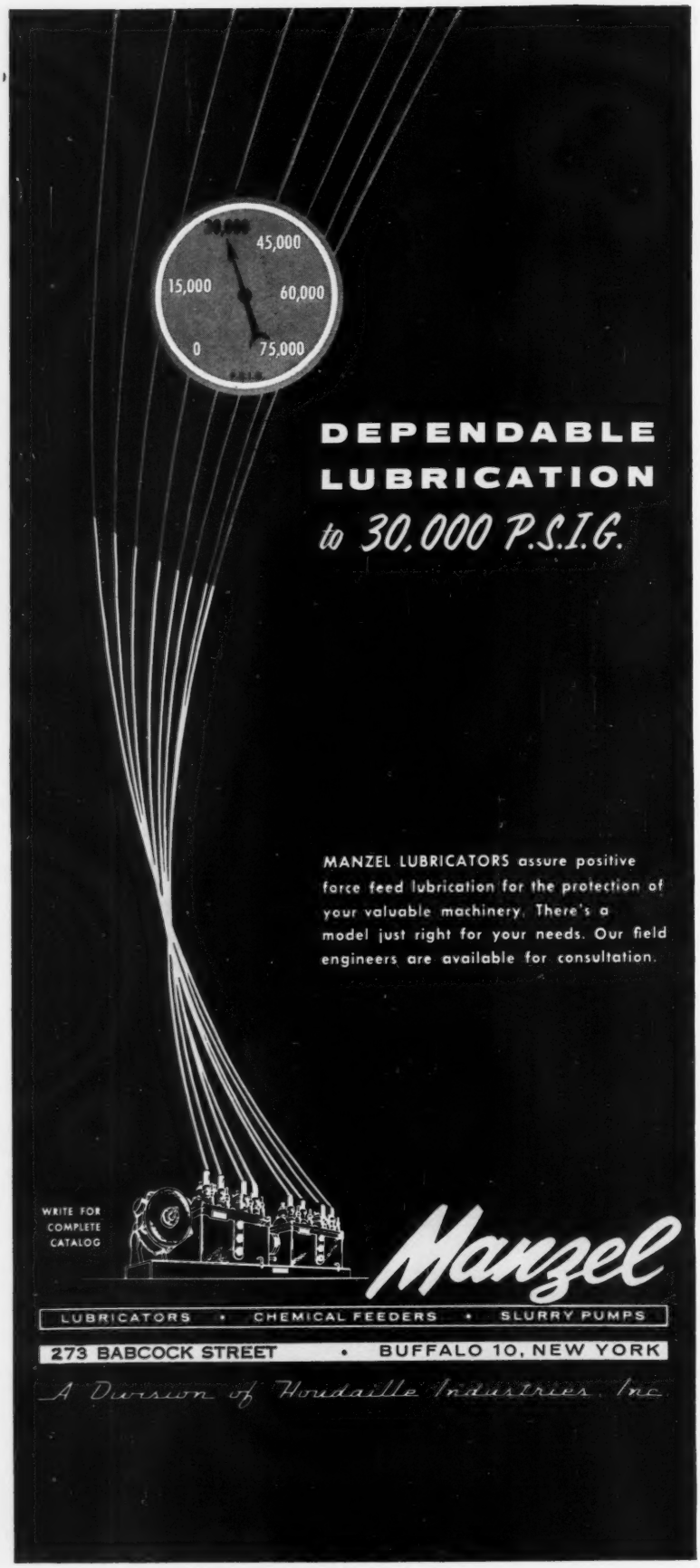
Bower engineering officials examine new bearings and assembly.

"Native" Alloy Steel Developed For Jet Engine Bearings

DETROIT—Jet engine bearings to operate at elevated temperatures are now made of a "99% native" alloy steel by the Bower Roller Bearing Division of Federal-Mogul-Bower Bearings, Inc., of Detroit. The material requires no imported components such as tungsten or cobalt.

The new steel is identified as Bower "315" and produced by Republic Steel Corporation. Its principal alloying elements are 3 per cent nickel, 1.5 per cent chromium and 5 per cent molybdenum.

The new bearing steel is currently used on the "hot" end of military jet engines. Operating temperatures range up to 500 degrees F., about twice the temperature for bearings used in internal combustion engines for automobiles and trucks. While steel ordinarily softens permanently at elevated temperatures, Bower "315" recovers its original hardness when cooled to room temperature after heating as high as 1000 degrees F.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LUBE IN NEWS

Battelle to Investigate High-Temp Bearings

To obtain design data and information on modes of failure for rolling-contact bearings operating at temperatures ranging from 300 F to 600 F, Battelle Institute has started a program of research for the Wright Air Development Center, Wright-Patterson Air Force Base, Dayton, Ohio.

Increasing temperature demands on airframe bearings in high-speed aircraft and guided missiles have pushed bearing requirements beyond the range of current design information. "Data are practically nonexistent for heavily loaded oscillating bearings in the 300 F to 1200 F range," according to Battelle technologist C. M. Allen, "and bearings and lubricants for airframe applications are not known which can be operated beyond 1200 F."

The present investigation at Battelle is the first phase of a three part program that has the ultimate objective of developing new bearings and lubricants and adequate design data for bearings to operate at these higher temperatures.

An evaluation of currently available bearings to obtain design data and information on modes of failure in the temperature range of 300 to 600 F constitutes the first phase. It will include a survey of a number of bearing producers and users to aid in determining some of the limitations and problems involved in the use of airframe antifriction bearings. Up to 1500 bearings may be evaluated in load-life experiments for the development of design data. In both the survey and the experimentation, a prime objective of this research for the Air Force will be to develop load-life data and an increased understanding of the mechanism of failure at moderately high temperatures.

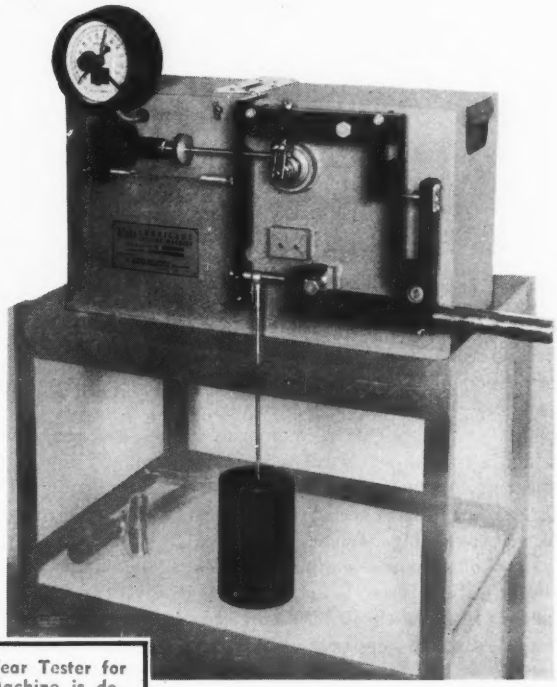
The mechanism of failure at elevated temperatures is not known except that above a certain temperature level, depending upon the material, surface fatigue may not be the dominating mode of failure. The failure, Allen says, might progress as a result of creep or plastic flow and wear.

The Better The Test Equipment...
THE LOWER THE COST OF PRODUCTIVE RESEARCH

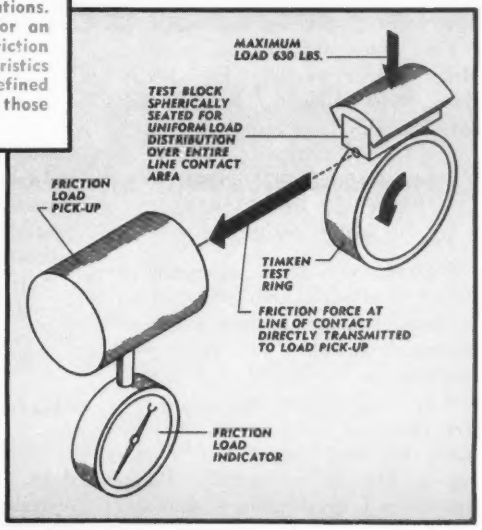
PRECISION TESTING IN THE LOW VELOCITY E. P. RANGE

Admittedly the greatest single handicap in boundary lubricant research has been the lack of tools in the form of accurate and reliable bench test equipment. The stark oversimplification of E.P. testing equipment contrasts sharply with the complexity of friction and wear phenomena. The Alpha Model LFW-1 is the result of a search for an "intermediate" between the oversimplified E.P. Bench Test and costly and time consuming full scale performance tests. Such intermediates, of which many are needed, cannot fail to sharply reduce the terrific waste of research expenditure and talent.

Since all E.P. lubricity ingredients function effectively only in a limited temperature range, E.P. Bench Tests must either measure frictional contact temperature or, pending the development of means to do so, simulate them. Sliding velocity is one important factor in this simulation. A large group of E.P. lubricity ingredients which are highly successful in industrial low velocity mechanisms completely fail to respond on most E.P. testers because excessive velocities and loads combine to produce frictional temperatures beyond the functional temperature range of the E.P. ingredient.



Alpha Model LFW-1 Lubricant-Friction-Wear Tester for sliding velocities below 43 feet per minute. Machine is designed for low velocity extreme bearing pressure applications. Developed by professional testing machine designers for an accuracy of better than 98% in load, velocity and friction measurements, results reflect the performance characteristics of a lubricant in a uniformly loaded and accurately defined friction area. The only variables from test to test are those intended by the operator.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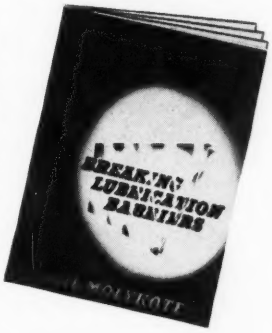

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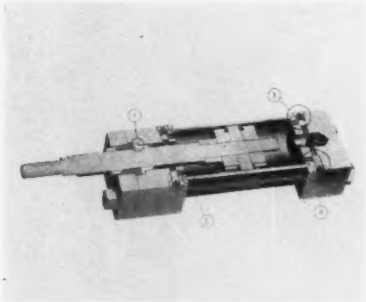

ings on high pressure hydraulic cylinders and permitting the introduction of a new line of medium to high pressure cylinders. A tube end seal of Teflon plastic and a rod bushing seal of the same material replace "O" ring seals formerly used which were subject to extrusion and shearing. A new cushion adjusting screw, flush with head or cap, has a self-regulating Teflon lock seal which holds the seal no matter where the adjusting screw is set. The redesigned ball check is now located directly opposite adjusting screw. If interference makes adjusting screw difficult to reach, it may be interchanged with the ball check.

For more information, contact Miller Fluid Power Div., Flick-Reedy Corp., 2040 N. Hawthorne Ave., Melrose Pk., Ill.

HYDRAULIC EQUIPMENT PROVIDES AUTOMATIC LUBRICATION

Hydraulically operated equipment such as continuous miners in the coal industry, shapers, surface grinders, planers, broaches, milling and boring machines in the machine tool field and hydraulic presses of all types can now receive the advantages of automatic centralized lubrication by adding a Trabon Hydra-Lube Pump through a Trabon feeder system and reservoir.

No special valves or drives are needed. The pump connects to the existing supply lines of the hydraulic system and guarantees positive lubrication as the machine operates under full load to all possible points of friction.

Hydra-Lube Pumps are available in three sizes and are designed to operate where the machine cylinder is hydraulically powered in either one or two directions. Lubricant (oil or grease) discharge is completely adjustable to the needs of the machine. Dual "O" ring seal with vent between eliminates possibility of mixing lubricant and hydraulic fluid. The pumps are designed to operate in conjunction with lubricant reservoir assemblies of 1½, 3, 5, 8, and 12 pound or pint capacities.

For further information on the Hydra-Lube Pump, write for Bulletin 5531, Trabon Engineering Corporation, 28815 Aurora Road, Solon, Ohio.

CORROSION-RESISTANT PUMP

An air-operated reciprocating pump, now available, is said to be tough enough to withstand the effects of corrosive materials.

The stainless steel pump will handle fluids ranging in temperature from —30° F to 220° F at discharge outlet pressures of 1½ times the incoming air pressure and is capable of operating on an air supply range of 30 to 200 psi and pumping from one to 20 gallons of fluid per minute, depending on the air pressure used. Material is delivered in equal amounts on both up and down strokes of the pump without pulsation.

The air motor operating the pump is entirely separate, making it impossible for fluid being pumped to mix with the air mechanism.

Further information can be obtained by writing Balcrank Inc., Disney St., Cincinnati 9, Ohio.

FELT SEALS PROTECT BEARINGS

Die-cut felt seals are used in Flexidyne, a dry type "fluid" drive. Using a steel shot flow-charge principle, impregnated felt seals in combination with a steel labyrinth reportedly simplify the heart of the drive. Technical and application data is available from The Felters Co., 210 South St., Boston 11, Mass.

BENTONE* 34

Grease Manufacturers and Distributors

THE ACME REFINING CO., Cleveland, Ohio
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ALLUBE CORP., Glendale, Calif.
THE AMERICAN LUBRICANTS CO., Dayton, Ohio
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UNITED PETROLEUM CORP., Omaha, Neb.
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THE WARREN REFINING AND CHEMICAL CO. Cleveland, Ohio
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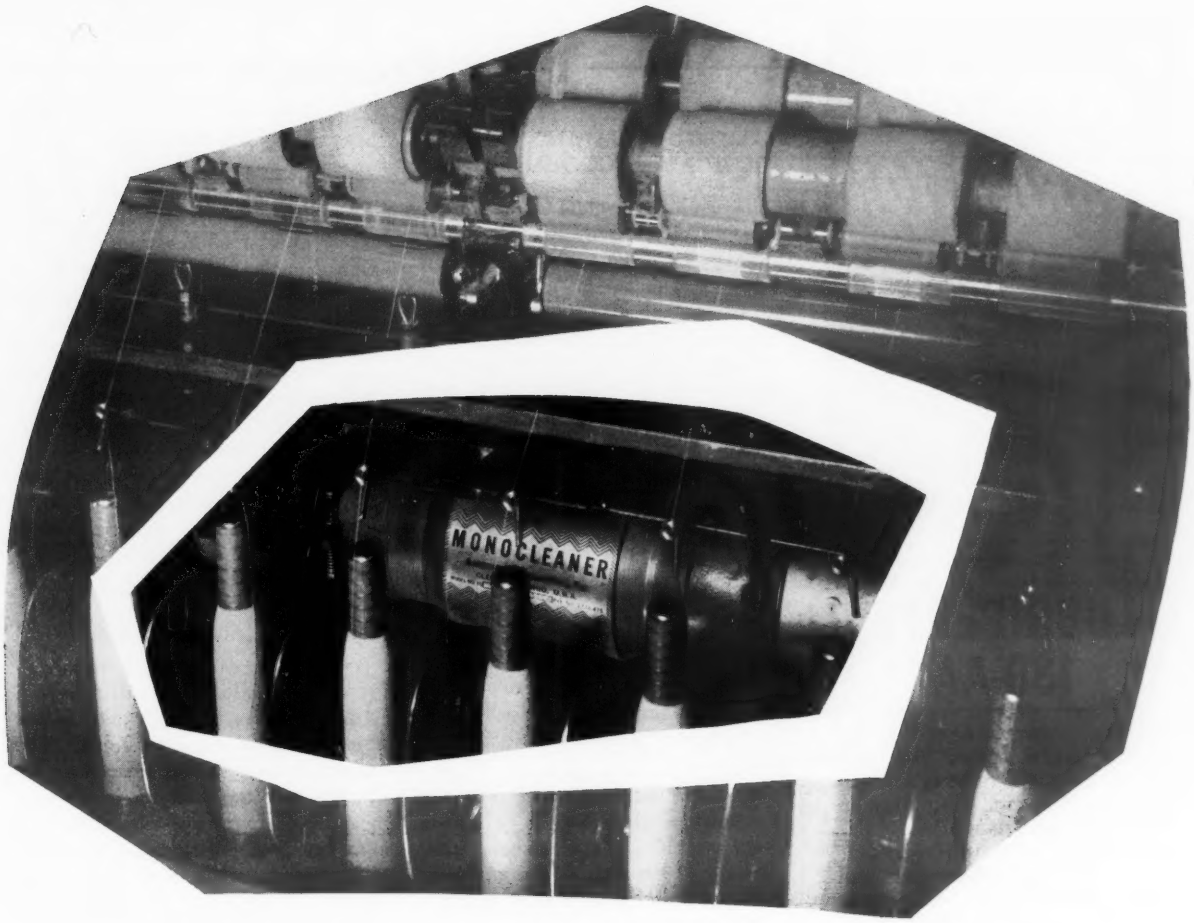
*T.M. Reg.

BENTONE·34®



For textile mills or

WHEREVER CLEANLINESS COUNTS

use non-melt grease made with BENTONE*34

High-performance grease made with Bentone* 34 is solving problems for textile mills, food plants, or any "problem installation" where a drop of grease can spoil a batch of products. The reason: *Grease made with Bentone 34 does not melt or bleed.* It stays in place under heat.

This has been proved by such firms as American Monorail Company, Cleveland, Ohio. Their lint cleaners for textile mills use grease compounded by using Bentone 34. Specify greases made with this remarkable non-soap, non-melt grease component. *Write for data or see your local supplier.*

Bentone* TRADEMARKS REGISTERED

BENTONE*34
FOR MULTI-PURPOSE GREASE

BAROID DIVISION
NATIONAL LEAD COMPANY
P. O. Box 1675
Houston 1, Texas

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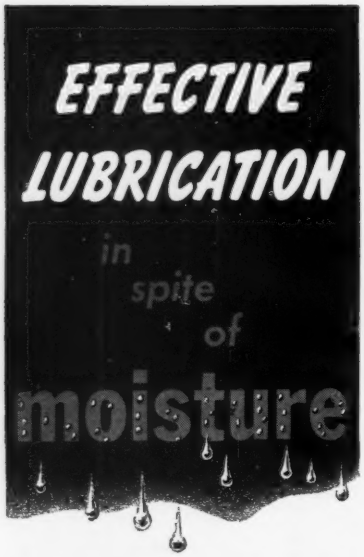


LUBRIPLATE 130-AA

affords machine parts and bearings real protection against rust and corrosion. It is a grease type lubricant that possesses the utmost in water and acid resistance. It has exceedingly high film strength. Due to the marked adhesiveness and water resistance of LUBRIPLATE 130-AA, it does not wash off readily, hence is the ideal lubricant in the presence of moisture.

REGARDLESS OF THE SIZE AND TYPE OF YOUR MACHINERY, LUBRIPLATE LUBRICANTS WILL IMPROVE ITS OPERATION AND REDUCE MAINTENANCE

For nearest LUBRIPLATE distributor see Classified Telephone Directory. Write for free "LUBRIPLATE DATA BOOK"... a valuable treatise on lubrication. LUBRIPLATE DIVISION, Fiske Brothers Refining Company, Newark 5, N. J. or Toledo 5, Ohio.

New Products

ARMORED TUBES AND NOZZLES PROTECT AIR, COOLANT LINES

"Trico-Mist" coolant systems, heretofore available only with Neoprene rubber protective tubes, are now equipped with flexible armored tubes to protect plastic air and coolant lines against abuse and damage. The flexible armored tube is provided with a substantial fitting which permits easy, quick replacement of the nozzle. Descriptive Broadside TMS-1 is available by writing Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wisc.

GEAR PUMP SEAL CUTS LEAKAGE

A sealed gear pump, measuring 2½" in diameter has been developed. Featuring a special seal, this unidirectional pump can be mounted on any machine requiring fluid circulation.

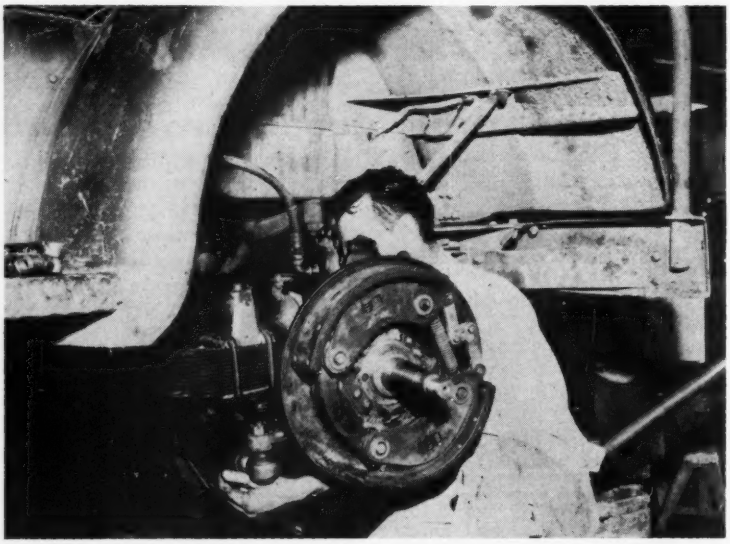
Intended for recirculation sump systems and designed to be driven from a rotating machine shaft either directly or through a gear or chain drive of suitable ratio, the pump is applicable to machine tools or to any other machine requiring "flood lubrication" over gear trains, chains, or cams.

The special lip-type seal, located on the drive shaft, is said to withstand back pressures up to 20 psi. Operational failure reportedly is eliminated by the venting of pressure to the pump inlet via a pressure relief system. A built-in relief valve prevents loading the seal above rated limitations. The pump can be driven in either direction, drive direction determining inlet and outlet ports.

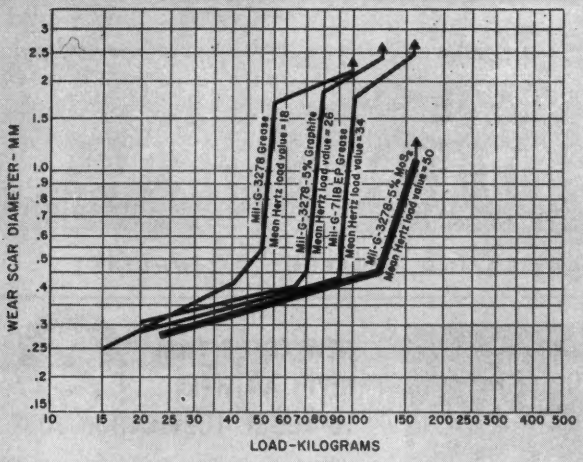
For more information about unidirectional sealed gear pumps, write or phone "Service," Bijur Lubricating Corp., 151 W. Passaic St., Rochelle Pk., N. J.

HANDLER CAN LIFT 800-LB. DRUM

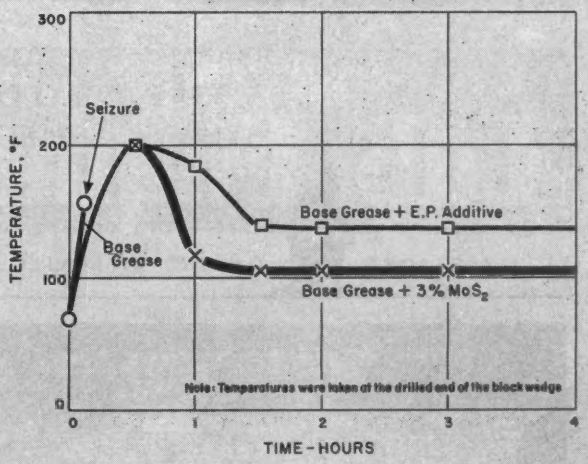
A one-man controlled, hydraulically operated lift which will elevate a drum 56 inches vertically to 63 inches for pouring has been developed. The drum is easily tilted, extending 10 inches over the tank lip in pouring position. Large wheels make the unit easy to handle for pouring, stacking, tiering, truck loading and unloading. For further information, write Hamilton Equipment Co., 31 Beckwith Ave., Paterson, N. J.



ADDITIVE ON THE ROAD: Front axle being reinstalled on truck, one of thirty vehicles being used in road tests at Southwest Research Institute, San Antonio, Texas, to evaluate the effectiveness of molybdenum sulfide additive in chassis grease. In these road tests, now under way, thirty vehicles — city busses, tractor trailers, and police cars — will travel a total of 1,500,000 to 2,225,000 miles.


WEAR SCAR DIAMETER - MM
LOAD-KILOGRAMS
Mil-G-3278 Grease
Mean Hertz load value = 18
Mil-G-3278 - 5% Graphite
Mean Hertz load value = 26
Mil-G-7119 EP Grease
Mean Hertz load value = 34
Mil-G-3278-5% MoS2
Mean Hertz load value = 50

4-ball E.P. test shows the superior extreme pressure properties of MoS_2 grease over the same grease without additives. In the same test, grease containing 5% graphite had a mean Hertz load value of 26; the conventional E.P. grease Hertz value was 34.


TEMPERATURE, °F
TIME - HOURS
Seizure
Base Grease
Base Grease + E.P. Additive
Base Grease + 3% MoS2
Note: Temperatures were taken at the drilled end of the block wedge

Timken endurance test clearly shows how MoS_2 functions when added to a base grease. With base grease alone, the specimen seized in about 10 minutes. With MoS_2 added, lubrication was effectively sustained throughout the test at a reduced temperature of about 100°, compared with a temperature of about 130° for the conventional E.P. grease.

Moly-Sulfide additive extends effective lubrication

Laboratory and field tests prove that chassis lubricants containing MoS_2 sustain lubrication after base grease is wiped off

When standard chassis greases are first applied to shackle pins, ball points and other bearing points under trucks, trailers and passenger cars, excellent results are obtained. These greases, however, are subject to mechanical motion, such as reciprocation, pressure, oscillation and shock loading which tend to eject the lubricant.

At this point, chassis grease containing 3% Moly-Sulfide proves its great value. MoS_2 is forced between the rubbing surfaces, adheres to the metal and forms a protective film that prevents galling, welding and fretting. The extent to which MoS_2 sustains lubrication is demonstrated by substantial reduction in wear.

Four tests show how MoS_2 reduces wear

1. **Shell 4-ball extreme pressure test** (see illustration) —MoS_2 addition reduces wear more than 50%, increases load capacity about 70% over base grease.

2. **Falex 2-hour wear test**—As the chart here shows, MoS_2 sustains lubrication, as proved by a reduction in wear, by an average factor of 500.

Grease	Loss in weight of specimen
Lithium-base	1,000 mg.
Lithium-base plus 2% MoS_2	1.98 mg.—average of 3 runs

3. **Timken endurance test** (see illustration)—MoS_2 sustains lubrication well beyond the capabilities of the base grease alone, and at a lower temperature than conventional E.P. grease.

4. **Simulated shackle bolt service test**—oscillating friction machine—3% MoS_2 addition virtually doubles the effective lubrication of a lithium base chassis grease —is far more effective than conventional additions.

Hours to failure—average of 2 tests

Additive	None	3% MoS_2	5% ZnO	15% ZnO
Dry	350	564	336	555
Wet	162	264	109	85

On-the-road use proves MoS_2 prevents chassis wear

A St. Louis company reports no noticeable wear on chassis parts of auto transport trailers, since they selected an MoS_2 chassis grease.

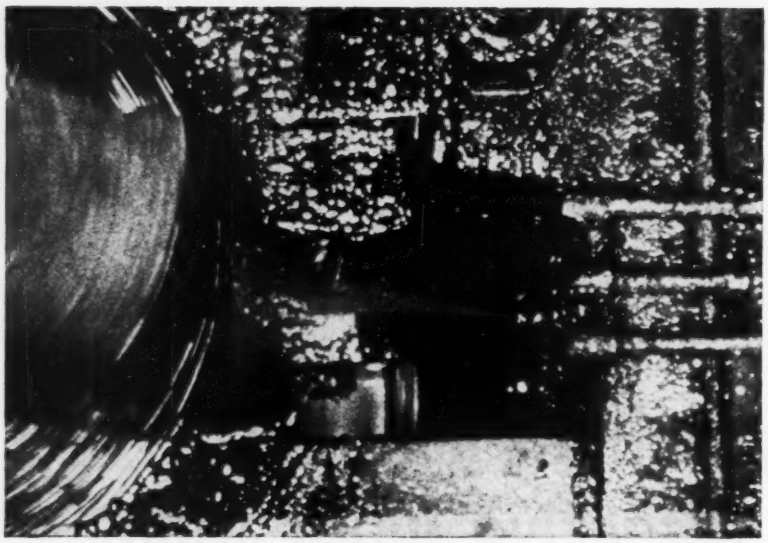
Based on their own tests, several major automobile and truck manufacturers use—and recommend—MoS_2 grease for torsion suspension units, ball joint suspension, as well as for many other critical-wear areas.

Moly-Sulfide greases are available from many producers today. For a list of the manufacturers of these greases—which includes several major oil companies—and for a copy of "Molybdenum Disulfide as a Grease Additive," write to Department 38, *Climax Molybdenum Company, 500 Fifth Avenue, New York 36, N. Y.*

THE BROOKS OIL CO.

Lub-Engineering News

A REPORT ON THE DIFFICULT LUBRICATING PROBLEMS OF INDUSTRY



SPRAY LUBRICATION OF ROLL NECK BEARINGS CUTS COSTS FIVE WAYS, STEEL MILL REPORTS

Lubrication of roll necks and bearings by means of a spray system employing a fluid lubricant, as shown in the unretouched photograph above, has proved far superior to the traditional system of forcing grease along grooves machined in the bearing surfaces, according to reports of an extended study recently completed by a major steel company.

Five major savings resulted from the use of the spray system:

1. The initial cost of preparing the bearings for service was reduced because it was not necessary to cut grease grooves. Also, the safe overall bearing area is substantially increased.
2. Lubricant consumption dropped from an average of 20 drums per week to approximately 3 drums.
3. Bearing life was substantially increased. With the old system of lubrication, bearings wore out in about 3 weeks. Bearings lubricated by the spray system showed no appreciable wear after 18 weeks of service.
4. Longer bearing and roll neck life also meant less time and cost for remachining and dressing.
5. Mill downtime was greatly reduced. With ordinary lubrication, 5 intervals of downtime would have been expected during the four months of the study period—with spray lubrication there was *no* downtime—saving thousands of dollars.

First adapted in 1949 and now being used by major steel companies for this important and severe application—the lubricant was Brook's Klingfast, a lead base fluid lubricant which retains its film strength (4 gram Timken test, 30 min. minimum—43 lbs. L.A.L.).

For complete data, write for Bulletin K-26-LE.

NEW COLOR FILM GIVES ANSWERS TO TOUGH LUBRICATION JOBS

Want to see how good lubricants get that way—lubricants for severe industrial applications, that is?

This new 30 minute 16 mm motion picture, in full color, with sound, shows a number of tough lubrication jobs. It pictures and describes the research, manufacturing and quality control methods that enable Brooks lubricants to "stand up" under severest service

It's available to any supervisory or engineering group interested in lubrication problems. No charge, of course —we're glad to have you see it. Just call or write any of our offices and tell them you want to borrow "The Brooks Oil Story."

HAVE YOU TRIED Klingfast® FOR OPEN GEARS?

Lubrication of open gears is one of industry's toughest jobs because of severe operating conditions in speed, vibration, shock, reversal, intermittent action, exposure to high temperature, contamination, and the added problems of inaccurate alignment, excessive clearance in bearings and incorrect contact of teeth surfaces. If you've been having open gear lubricating problems, write us about Klingfast. This lubricant has been the answer in many case studies.

BROOKS LABORATORIES ARE AT YOUR SERVICE

Extensive research is constant with Brooks. Our laboratories are available at all times, with experienced and qualified personnel to serve your lubrication research and development needs.

THE BROOKS OIL CO. 81st YEAR 1876 1957

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ASPERITIES... *a page of editorial comment and opinion*

Why Industrial Membership?

The values of established lubrication programs within Industry have repeatedly demonstrated the savings available to Industry through productive maintenance administered by the trained lubrication engineer. Industry's role in the affairs of ASLE, however, exceeds this fundamental recognition of lubrication engineering's worth. Industrial membership in the Society constitutes active participation in the research goals defined by Industry, the Technical Committees and Industry Councils of the Society *and* financial assistance in attaining these goals.

The Industrial member pays $150.00 each year to support the technical research activities undertaken by the Society. These activities include the accumulation and dissemination to the profession and to Industry of authoritative scientific information available through research, testing, field experience, special study conducted by various organizations relating to development and application of lubricants, machine design, and administration of related duties within the sphere of lubrication engineering activities. Higher standards of practice together with greater productive efficiency of basic products through understanding industrial production problems and their solutions is the ultimate goal of these activities.

To augment technical activities and studies undertaken by private firms, educational institutions, and research groups, the Society has established a research fund to stimulate investigation impossible within Industry. Policies of the Society provide for annual allocation of a portion of the Industrial member's fees to this fund.

Funds thus allocated are being used in the following manner:

- To define literature, research, and development activities essential to increased productivity.
- Preliminary discussions have been held with experts from two universities, and a Society-sponsored research project on soluble oils appears feasible.
- Technical Committees and Industry Councils of the Society are preparing a list of projects for sponsored research of great interest and potential value to Industry.
- A procedure has been established for financing sponsored research projects with funds in excess of those allocated from Industrial membership fees.

The first results of these funds are apparent. An extensive review of interrelated variables involved in the design, manufacture, and effective use of journal bearing elements will be released later this year by the Society. The project has been designed to provide a basis for which further analytical and experimental investigations could be determined and planned.

The resulting publication provides an authoritative survey of 2500 screened and abstracted references. Of the original 2500 sources, 563 references have been selected for inclusion in the book, reclassified on the basis of subject matter, and organized for quick access and reference. Emphasis has been placed upon the type of information available through the source and the manner in which it can be utilized most fully.

The author, Prof. D. D. Fuller of Columbia University's Mechanical Engineering Department and principal scientist at Franklin Institute Laboratories for Research and Development, has provided a detailed introduction to journal bearing design and development as well as a conclusion section which recommends further analytical and experimental investigations needed to increase basic understanding of design and operation of journal bearings.

Industrial participation through Industrial membership affiliation in future projects of this nature will provide additional information essential to both Industry and to the profession. Has your company considered Industrial membership?

J. O. McLEAN, President
American Society of Lubrication Engineers

AT A CHRYSLER AUTOMATED ENGINE PLANT

Peak Output Through Planned Lubrication

by D. F. Miller, Chrysler Corp.

These unique problems of mechanized industry:

- ***More Key Points to Lubricate***
- ***Critical Metering of Delivery***
- ***Elimination of Downtime***

Indicate: Engineered Lubrication — a basic building block in the automation pyramid

THE first function of the lubrication engineer is to convince management of the benefits to be derived from lubrication engineering practices. This has been particularly true of industrial lubrication in the automotive industry.

With the increasing use of automation, where the failure of a single machine can cause widespread work stoppage, the need for proper lubrication practices is readily demonstrated. Management now requests the best technical practices available —Lubrication programs are no longer forgotten recommendations; they begin to take shape at the time new plant layouts are begun.

Automation, the optimum use of personnel and machines to produce large quantities of high quality product at minimum cost, implies operation without breakdown, long machine life, good operational efficiency and other factors that assure continuity of operation. This requires the selection of optimum lubricants and procedures governing their use.

The objectives of the lubrication engineer in an automated production facility can be defined broadly:

1. Reduction of costs of machine operation
2. Insuring continuity of machine operation
3. Prevention of costs resulting from machine repairs.

His activities in attaining these objectives are, of course, allied with the selection and use of the most suitable lubricants for specific applications.

SETTING UP A PROGRAM

Establishing lubrication practices in a modern plant with automatic handling equipment will naturally vary with particular conditions, but the following basics should apply to virtually any production facility.

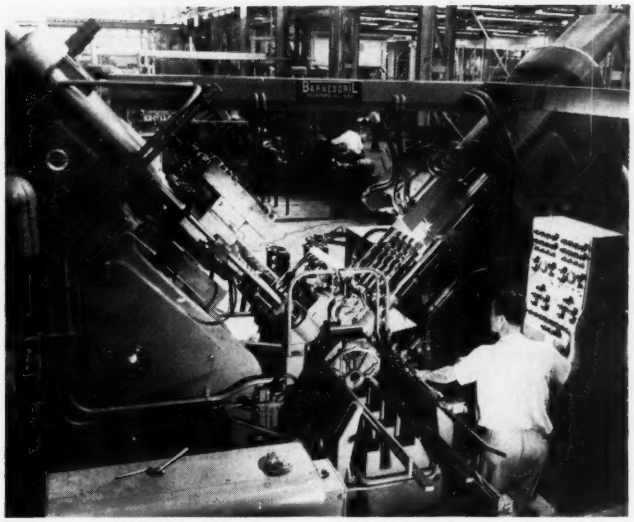
a—The selection of the proper lubricant for each application.

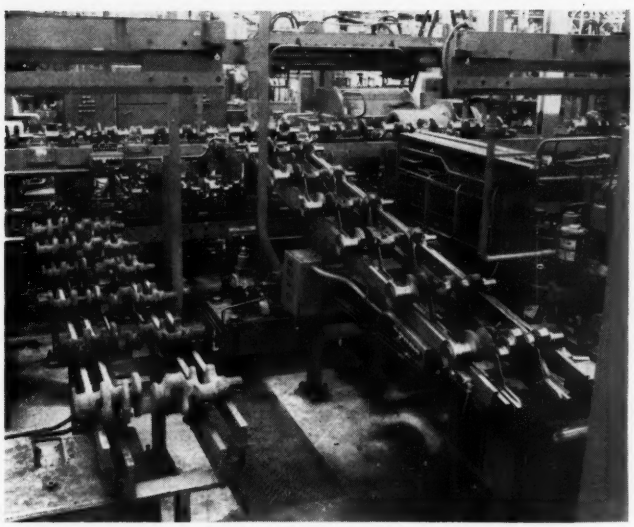
b—A minimum number of lubricant types to provide minimum handling costs, optimum purchase price, and reduction of the possibility of error in handling.

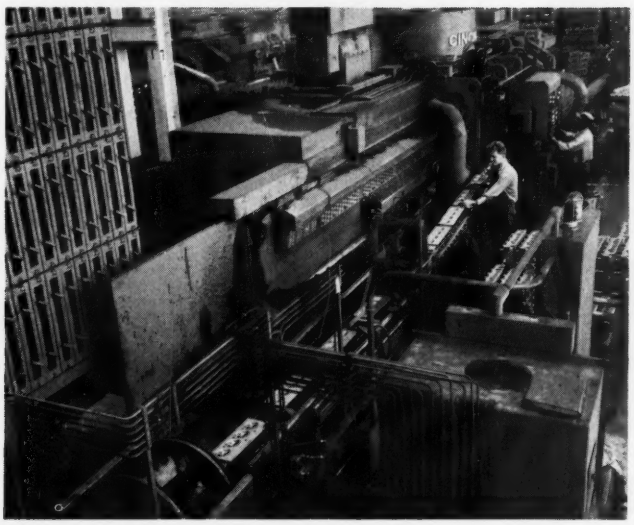
c—The proper handling, storage, and dispensing of materials, insuring the proper quantity of lubricant applied at the correct interval in uncontaminated condition.

d—Effective schedules and record systems which will alert personnel to incorrect conditions, and predict the usable life of lubricants and machines.

e—Effective purchase control to maintain the necessary lev-







Opposite page: Control bridge located over four cylinder head lines to enable operator to control major sections of each of two lines and banks serving the lines.
Top left: Engine blocks in honing machine used for finishing cylinder bores. Each 1380 ft. long cylinder block line contains 70 machines — broaches, assembly machines, pressure testing machines, qualifying fixtures, and numerous automatic gaging stations.
Center: Walking beam in automation crankshaft section. Beam feeds crankshafts downward to grinding machines.
Bottom: Overhead view of cylinder head branch.

el of quality as established, to take advantage of improvements available and to avoid unnecessary deviation from standardization.

f—Sufficient number of trained personnel to carry out the program.

g—Confidence of the using personnel and machine operators, as well as management, in the effectiveness and desirability of the lubrication program.

The selection of proper lubricating materials, based on cost-performance, is a continuous problem facing the lubrication engineer in any plant. The selection of material is often complicated by the number of dissimilar applications which do not justify a special additional lubricant or an extensive program of evaluation.

For example, a production plant may have machines varying in type and size from high speed, air driven grinders which weigh a few ounces, to large presses which occupy hundreds of feet of floor space. The selection of lubricants then becomes a problem of establishing the optimum, not only for a single machine type, but in terms of the entire lubrication program.

The use of automation has influenced lubricant selection in several ways. Because standardization of machines is necessary for automation processes, more extensive planning and coordination is required in machine selection and design. More consideration to lubrication requirements is given at a point early in the development of machines involved in the

processes. Greater emphasis during the planning stage is placed on the standardization of lubricants which require automatic dispensing and control.

In many organizations, a lubrication committee has been found to be effective in maintaining the necessary degree of purchasing control and standardization. In its smallest form, this committee may consist of only the lubrication engineer and the purchasing agent.

BENEFITS FROM STANDARDIZATION

The necessity for standardization of materials and procedures of all types is inherent in automation because of the inter-dependence of machinery. There are many additional benefits to be derived from lubricant standardization. Since initial costs of materials are usually related to the volume of purchases, it is obvious that a purchase price advantage can be obtained through reducing the number of different products purchased, which, in turn, results in larger quantities of standard materials. However, performance losses, though perhaps eventually more costly, are often less quickly recognizable than are savings in lubricant purchases.

Quantity purchases resulting from standardization may also reflect cost reduction as well as improvements in the storage and handling of lubricants. The use of large central storage tanks and piped systems of supply to local areas will not only reduce manual handling costs but will materially aid in providing uncontaminated lubricants to the machine in question.

The cost of piping facilities and filtering devices which are frequently not used with individual storage and dispersing units can be more readily justified for equipment handling large volumes of materials. In terms of lubricant contamination, however, especially where non-standard items are used, it may be advantageous to purchase lubricants in very small units which will reduce the amount of contamination in containers which are repeatedly opened to dispense lubricants. Lubricant suitability is in this way related to the volume of purchase. Effective communications with the purchasing department are usually necessary in cases where very small unit purchases are desired.

Standards which result in fewer materials reduce the possibility of errors in lubricants supplied to the machine. Multiplicity of lubricant application, necessary in automation, and inherent in standardized selection of lubricants, greatly increases the likelihood of the proper lubricant reaching all necessary points. This is further improved through the use of standardized and simplified dispensing equipment.

Because automation machinery is more complex than individual machines, a greater number of points requiring lubrication can be expected. In addition, a larger proportion of these will be less readily accessible to the personnel assigned to the lubrication of the machine. In many instances, the proper quantity of lubricant is more critical in automation machinery. *In all cases, the positive assurance of adequate lubricant to provide continuous operation is more important in automation equipment.* These factors all indicate the necessity of using automatic dispensing equipment, the selection of lubricants suitable for use in automatic dispensers and the proper scheduling of relubrication. These are important points in any program of lubrication.

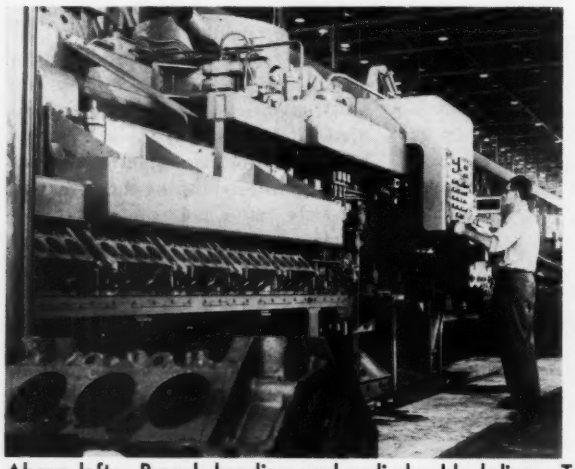
RECORDS

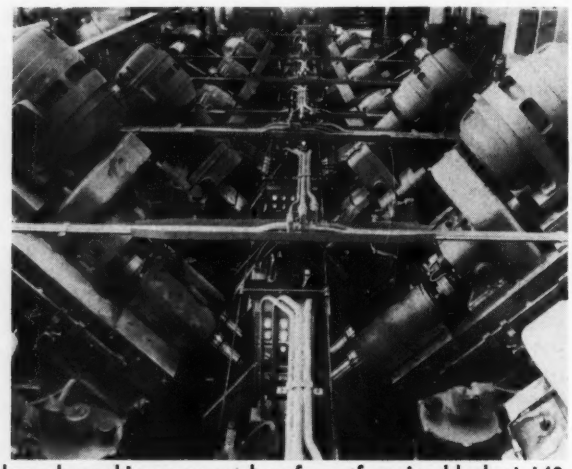
The necessity for continuous operation with a minimum of breakdown greatly increases the demand for the maintenance of records on lubricant consumption, machine life and quality of the machine operation. Automatic machinery very often incorporates devices which alert the operator to changes in the machine itself, such as tool wear. Optimum use of machines cannot be obtained without adequate maintenance records which will indicate the departure of the machine from optimum operation.

The greater dependence of automation equipment on effective preventive maintenance possible through the programs of suitable lubrication has provided the need for greater numbers of personnel adequately trained in lubrication understanding and application.

THE LUBRICATION ENGINEER'S ROLE

The specific activities of the lubrication engineer in any production plant will vary with the organization of responsibilities, the type of end product, the physical nature of the plant, and the attitude of management toward lubrication and pre-





Above left: Broach heading each cylinder block line. Two broach machines now work surface of engine block at 140 per hour rate. Above right: Blocks in section of line where cylinder bores are semi-finished. An alternate boring pattern is used to allow for large diameter heavy duty spindles necessary to attain precision.

ventive maintenance while the practices are being established. In illustrating what may be required, the following description presents briefly the activities of personnel concerned with lubrication in an automated engine plant during the planning stages and early production period.

After the basic scope of manufacture to be performed was established, machining operations were studied to determine the necessary metalworking fluids to be used. This selection of fluids and required amounts became a part of the information necessary to determine the storage space needed, types of filtering and fluid handling equipment, methods of chip removal and safety devices necessary for control and handling of metalworking fluids.

As machine tools and other machinery were decided upon, preliminary discussions regarding lubrication requirements were held with machine builders to establish types of lubricants, probable quantities, types of filtered mechanisms and dispensing equipment to be used on purchased machines. Further studies were necessary to establish storage facilities required, types of filters, and oil and grease dispensing equipment to be used.

Space requirements and storage location for fluid lubricants and hydraulic oils were determined.

During the period of machine installation the final selections of lubricants were made with consideration given to machine builders recommendations, past experience with similar machines in older plants wherever applicable, and the standardization of materials in methods already established. Simultaneously, instructions were prepared and distributed to personnel responsible for procurement, handling and application of lubricants. Tentative schedules for relubrication were prepared subject to revision as experience would require. Information regarding standard materials and procedures for quality control inspection of incoming shipments and control of metalworking fluids were prepared and distributed as necessary to personnel assigned to these functions.

During the very early period of production, some compromises were necessary in lubricant selection and procedural standardization in order to expedite actual machine operation. As production became stabilized and details of operation became more familiar to operating and maintenance personnel, greater emphasis was again shifted to standardization and the establishing of optimum practices and materials for all operations. Maintenance records began to provide further information to indicate where improvements were necessary.

The activities described were carried out over a period of approximately one year. The program of lubrication improvement is a continuous one, however, and can be expected to continue throughout the production life of this plant.

As industry moves toward the greater use of automation, good lubrication practices will become a more significant part of production engineering. The future of lubrication engineering in the automotive industry becomes more encouraging as changes in lubrication policy are made with each new production plant which includes automation equipment.

ERRATUM

Through an oversight, Fig. 5 of the article "Specific Heats of Aircraft Engine Lubricants at Low Temperatures" by Schrock and Gott, which appeared in the July LE, was duplicated. The correct graph for Figure 6 is presented at right. We regret any inconvenience this may have caused our readers.

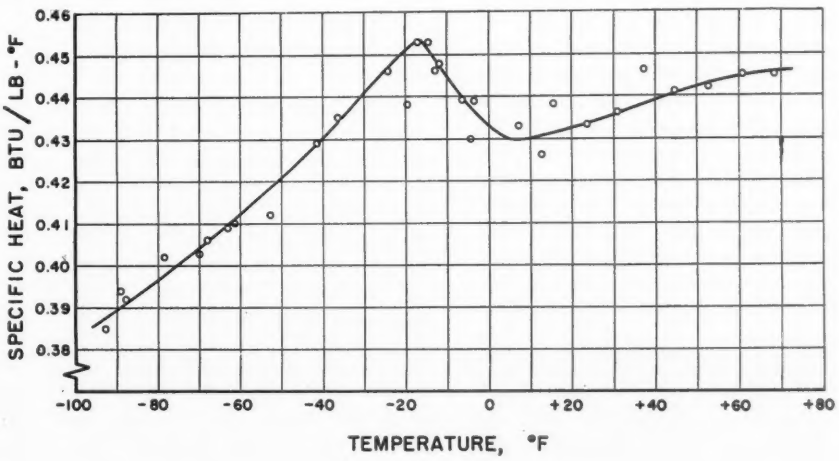


FIG. 6 SPECIFIC HEAT OF TURBINE ENGINE LUBRICANT

It's Toronto in '57

ASLE-ASME 4th Lubrication Conference

October 7-8-9 Royal York Hotel

Lubricating Oil Filtration on Diesel Engine Railroads

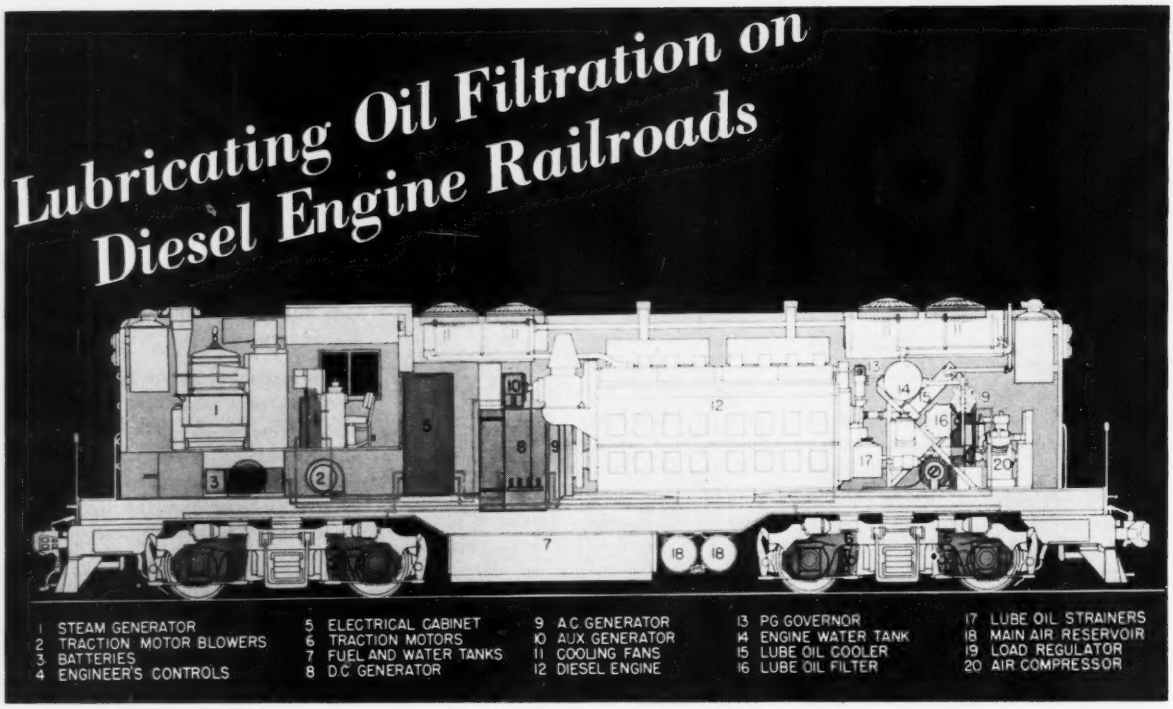


Fig. 1 A typical locomotive layout. Type shown is a GP-7 General Motors road-switcher.

by W. King Simpson
Technical Director, Fuels and Lubricants
General Motors Corp., Electro-Motive Division

ONE of the most important items which controls the successful operation of our diesel engines in railroad service is the filtration of the engine oil. Despite the fact that we, as an engine and locomotive builder, have emphasized the importance of oil filtration, lubricating oil performance is a neglected item on a few railroads. Surprisingly enough, this condition exists on railroads which in all other respects would be classified as being a good operating railroad. Poor quality filters and extended filter life beyond reason have an immediate and serious effect on our engine operation.

FILTER PERFORMANCE FACTORS

The history of lube oil filter performance through the years has indicated that the filter itself has been given a more difficult assignment with every successive change. In the early days, straight mineral oils were easier to filter and it was quite simple to evaluate filter performance on these oils as compared to the present day heavy-duty type products which are required by the higher output engines. Also, lube oil oxidation is accelerated at higher operating temperatures and this affects filter performance adversely. Higher sulfur content fuels and other degraded fuel properties directly affect oil filter performance and life. Higher additive level oils disperse the contaminants so effectively that the filter cannot hold them, thus defeating the purpose of the filter.

At the same time, there are extenuating circumstances which make the lube oil filter situation somewhat more difficult to control on the railroads than would be encountered in other industries. Figure 1 is a typical layout of our general-purpose type locomotive. Notice that the arrangement of the locomotive is such that there is very little available space. Operating under full load, the engine draws over 5,000 cubic feet of air per minute, which means that under severe weather conditions the oil will be very viscous and generous provisions for by-passing oil filters must be provided. Figure 2 shows a closer view of the engine itself, illustrating the arrangement of the lubricating oil system of the engine. The positive-displacement scavenging pump draws oil from the oil pan through the fine mesh screens and forces it through the lub oil filter and oil cooler assemblies. From the oil cooler, the oil is delivered to the oil strainer assembly where it is ready for recirculation by the combination piston cooling and oil pump.

Since the scavenging oil pump draws a greater quantity of oil through the strainer than is required by the lubricating oil and piston cooling pump, the excess oil returns to the oil filter sump. A relief valve is built into the filter housing in order to allow the passage of oil to the strainer housing in excess of the capacity of the oil filter elements. From this it is evident that the arrangement, construction and widely varying temperatures encountered in a locomotive dictate the use of a by-pass filter system.

Figure 3 shows our standard four-element lubricating oil filter housing with the filter containers in place. This type of housing is employed in all of our general-purpose and freight-type locomotives.

FILTER TYPES

In the early days of diesel locomotives, these canisters were packed with cotton waste, approximately six pounds of waste being jammed in each container. Shortly afterward, the various filter cartridge manufacturers started to supply filter socks which had their own center tube and screen and which were made to fit the original filter housings. We

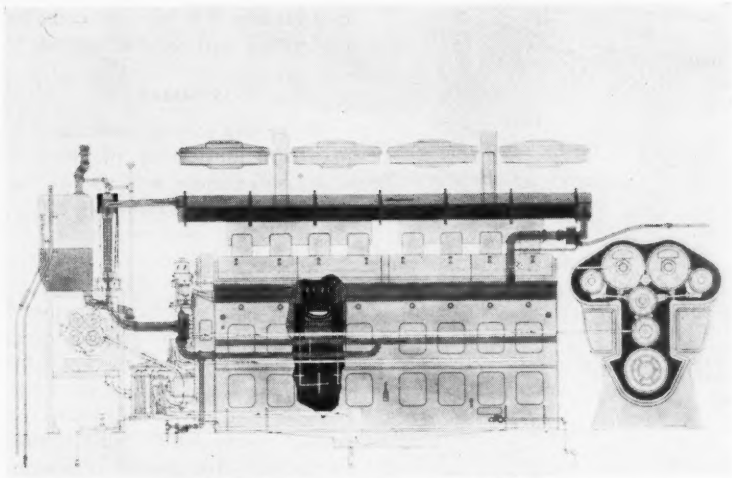


Fig. 2 Schematic diagram of Diesel Engine.

started a research program on diesel engine lubricating oil filters and their performance years ago and this project is still active. Two basic types of filters were investigated thoroughly; these might be classified as surface-type filters vs. depth-type filters.

It is our opinion that the surface-type filters were not satisfactory because they tended to build a layer of sludge and this build-up of the filter cake resulted in premature filter plugging. The summation of all this work has lead us to the conclusion that for the best all-around filter performance, the use of cotton waste in a properly constructed cartridge is the best selection that can be made to fit the idiosyncrasies of diesel engine operation on the railroad. Along with our research program on diesel engine lube oil filtration in the field, laboratory bench test data were obtained using the type of equipment designed by the lube oil filter test committee. Both standardized dust and standard oil sludge test procedures were employed and the results indicated that surface-type filters were much more efficient than depth-type filters. Despite this conclusion, the diesel engine indicated an outstanding preference for cotton waste, a depth-type filter.

FILTER PERFORMANCE

Quite a bit of detective work may be accomplished by occasionally inspecting the condition of the used oil filters when they are removed from service. Figures 4 and 5 are characteristic of outstanding disadvantages encountered in filter operation. Figure 4 shows separation or "doughnutting," due to the fact that the cotton waste was not properly placed down the length of the filter sock. Figure 5 is typical of what will happen when filter elements are used for too long a period of time. This element, unfortunately, was in one of our diesel engines for three months. The ordinary filter change period for freight operation would be 5,000 miles, which is equivalent to about two weeks of operation. When the cotton waste filters are neglected this badly, the individual strands lose their strength and disintegrate. This, of course, could cause a very serious engine failure.

The normal condition of a used cotton waste filter after two weeks' operation should find the filter element in good physical condition without any sludging on the outside of the element. If sludge is built up on the outside, the lubricating oil itself has been in extremely poor condition and the engine may have a water leak. In our experience, any build-up of deposit on the outside is due to some abnormal engine condition.

Most of the railroads have some correlation based on used oil properties which indicate whether or not the lube oil filter is performing satisfactorily. Using oils of relatively low dispersancy level, even the blotter test spots may be used to very good advantage. On oils of this type, if the benzene insolubles are kept below 0.10%, the filter elements in our engines are performing satisfactorily. Unfortunately, when higher additive level oils are used, or when chrome-plated rings or chrome-plated liners are used, the performance of the lube oil filter element is much more difficult to evaluate by laboratory control tests. It is also evident from spectrographic analysis of our used oils for silicon content, that lube oil filter practices cannot control high silicon values. If silicon is over 20 to 25 ppm in the used oil, the efficiency of air filtration must be improved or engine wear will become accelerated. It is not feasible to reduce the high silicon content of the used oil by lube oil filtration alone.

Analyses show that the contaminants found in our used lube oil filters have about the same general composition: about 12 to 16% is oxidized oil (pentane insoluble), about 2.5 to 5% is resins (benzene insoluble), about 97 to 99% is volatile while the remainder is ash consisting of 1 to 3% by weight. The ash consists mainly of the additive metals being used and all of the engine wear metals plus air-borne contaminants.

To anticipate the question that might be asked, as to what happens to our engine when no lube oil filter is used at all, we have a rather surprising answer. In a very short time we encounter difficulty from erosion of our main and connecting rod bearings. This is the first and unusual effect of not employing any filters at all. For those who are not aware of the situation, the performance of our main and connecting rod bearings is normally very good and usually the overlay on the bearings remains in good condition for three or more years. In contrast, without filters, the overlay will be eroded in several months' time. From this, we might assume that the by-pass filtration system must be adequate to protect all of the main and connecting rod bear-

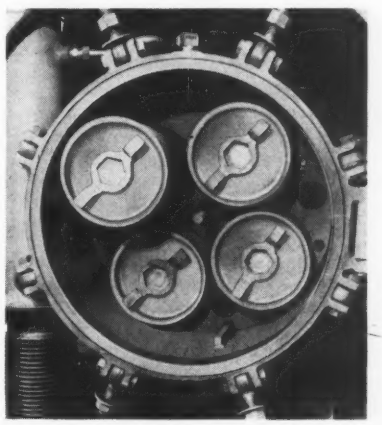
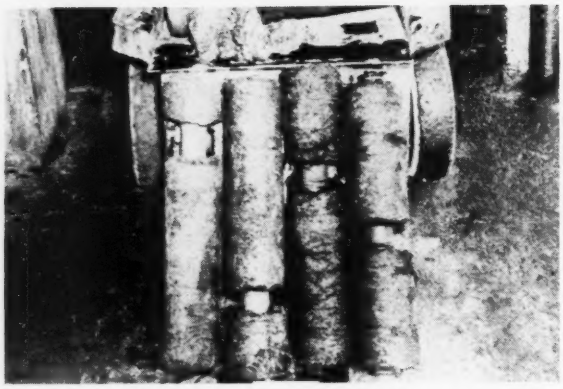
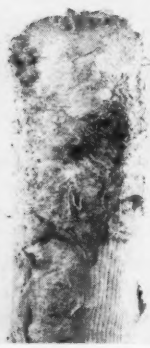


Fig. 3 Filter housing with filter elements in place.





These illustrations graphically depict the effects of usage on diesel engine filters. Fig. 4 (above) shows the separation of the filter element after long, though not undue use, while Fig. 5 (left) demonstrates the result of allowing a filter to remain too long in service.

ings because they are one of the best performing parts of our engine.

SUMMARY

We believe that a conservative approach to lubricating oil filters and their performance level is the most satisfactory situation as far as good diesel engine performance is concerned. We believe that a program to save money on lube oil filtration is a very poor way to accomplish the desired result. The size and shape of the locomotive unit, as well as the diesel engine operating conditions, have dictated the type of lube oil filtration system which must be used. Fortunately, the system is adequate as long as the construction and materials used in the filter elements are satisfactory and provided the filter change frequency is controlled in a reasonable fashion. The control of oil filtration is not too difficult, especially if a conservative condemning limit is employed. The performance of the lube oil filter is so important to the engine that it can improve engine performance to a remarkable degree with a substantial saving to the railroad.

Patent Abstracts

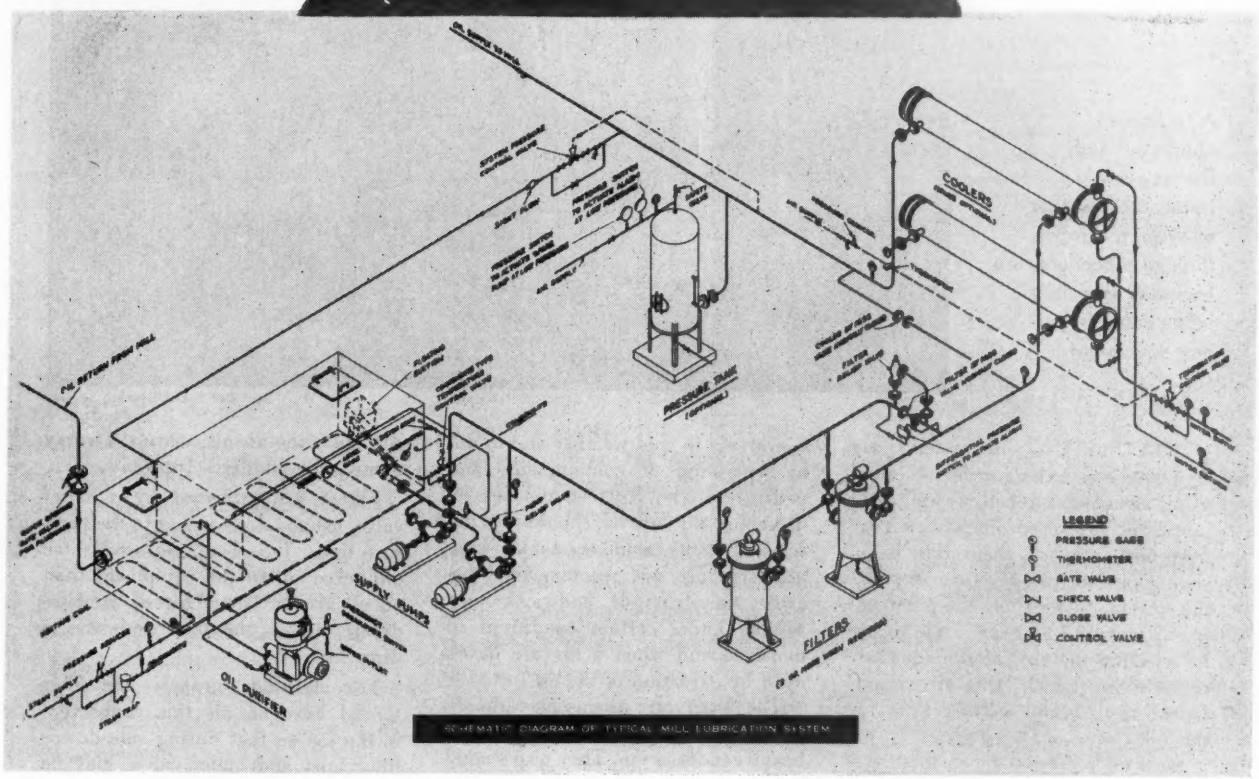
(Compiled by Ann Burchick, Aluminum Co. of America.)

Soluble Oil, Patent No. 2,770,598 (J. L. Jezl, assignor to Sun Oil Co.) A soluble oil composition comprising: mineral lubricating oil; alkali metal soap of petroleum mahogany sulfonic acids; alkali metal soap of petroleum naphthenic acids; alkali metal soap of carboxylic acids obtained by partial oxidation of mineral oil; 2.25 to 5.0 volume percent of water; and 1.25 to 3.0 volume percent of a compound selected from the group consisting of compounds having the formula $RCH_2O(CH_2CH_2O)_nH$ and compounds having the formula $RCOO(CH_2CH_2O)_nH$, R in each formula being a radical having 4 to 21 carbon atoms inclusive and selected from the group consisting of aliphatic and cycloaliphatic radicals, and $_n$ in each formula being an integer greater than 3 and within the range from $(x - 4)$ to $(x + 4)$ where x is the number of carbon atoms in R; said soluble oil having sulfonate saponification number equivalent within the range from 3 to 10 mg. of KOH per gram and carboxylate saponification number equivalent within the range from 10 to 20 mg. of KOH per gram.

Anti-Rust Oils, Patent No. 2,771,371 (E. A. Dieman & P. H. Ravenscroft, assignors to Standard Oil Co.) A composition for preventing rust formation on ferrous metal surfaces which are normally exposed to atmospheric air and moisture, which composition consists essentially of a phenol extract of a petroleum oil of lubricating oil viscosity, from about 1% to about 30% of mahogany soap, and from about 1% to about 10% of a mixture of saturated aliphatic oxygen containing compounds including alkyl esters, lactones, ketones, alcohols, and alcohol ketones, said saturated aliphatic oxygen containing compounds being obtained by the partial oxidation of petroleum hydrocarbons.

Polymethylpolysiloxane-Thickened Phenylmethyl Silicone Oil, Patent No. 2,771,422 (G. V. Browning & C. G. Brannen, assignors to Standard Oil Co.) The method of preparing a homogeneous thermally-stable organosilicone polymer grease consisting essentially of a thermally stable organosilicone polymer having a viscosity in the lubricating oil range of from about 10 to about 600 centistokes at 100°F. thickened to grease consistency with a gelled polymethylpolysiloxane, comprising mixing from about 40% to 98% of a liquid polymethylpolysiloxane substantially free of unreacted — OH groups with from about 2% to about 60% of a liquid organo-silicone polymer, other than said polymethylpolysiloxane, thermally stable at temperatures up to about 550°F., and having a viscosity in the lubricating oil range of from about 10 to about 600 centistokes at 100°F., heating said mixture to a temperature in the range of from about 400 to about 550°F. for a period of from about 1 hour to about 72 hours to form a gelled mixture, and milling the resultant gelled mixture with a sufficient amount of a thermally stable organo-silicone polymer having a viscosity in the lubricating oil viscosity range of from about 10 to about 600 centistokes at 100°F. to form a homogeneous grease containing from about 10% to about 80% of said gelled mixture.

Extreme-Pressure Lubricants Containing Diesters of Chlorendic Acid, Patent No. 2,771,423 (A. Dorinson, assignor to Sinclair Refining Co.) A metal-working oil composition consisting essentially of mineral oil and a diester of chlorendic acid in which the ester groups contain from 4 to 8 carbon atoms; the amount of diester present constituting about 10 to 75 percent of the composition by weight and being sufficient to impart extreme pressure properties to the composition.



SCHEMATIC DIAGRAM OF TYPICAL MILL LUBRICATION SYSTEM

Circulating Oil Systems:

Have you missed any of these points in check-out:

A PRACTICAL MANUAL

by R. E. Forsythe
Asst. Lubrication Supervisor,
The Steel Co. of Canada.

- ☐ 1. Before starting the system, check all feed and return lines to all pieces of equipment.
- ☐ 2. Open valves to by-pass the cooler in "cold weather."
- ☐ 3. Start the pumps. It is not necessary to close valves on the tank during shut-downs. The oil will not pass the pumps and there always is the risk of someone starting the pumps without first opening the valves.
- ☐ 4. Turn the filter switch to automatic.
- ☐ 5. Adjust the oil level in the pressure tank.
- ☐ 6. Re-check all lines to equipment looking for leaks.
- ☐ 7. Switch on the signal or alarm system.
- ☐ 8. Re-adjust the pressure tank.
- ☐ 9. Mark the operating oil level on the reservoir oil level indicator.
- ☐ 10. Adjust the oil going through the cooler to the correct operating temperature providing that the return oil is warm coming back, indicating gear cases and return lines are all warmed and returning oil properly.

Gulf Petro-Engineering Service

A technician checks out and flushes a newly-installed circulating oil system. Proper flushing procedures are essential, especially in the case of new equipment.

CIRCULATING oil systems are employed where large volumes of oil are needed to cool as well as to lubricate. More specifically, a gear transmitting more than 100 h. p. should be lubricated with a circulating system. Fifty times the pump capacity in gpm is a good guide to use in selecting the size of the reservoir or receiving tank. This allows sufficient resting and settling time for the oil.

Steel mills use pumps on these systems with capacities of from 5 gpm to 500 gpm which develop pressures up to 70 psi. Pressure is required to force the oil through filters, coolers, feed lines and sprays to the gears and other parts. Pressure may be controlled by pump speed, relief valves, different sized orifices at bearings and sprays, temperature of the oil and pressure control valves.

Two storage tanks or reservoirs are preferable in the modern circulating system. Many of these tanks are divided into a return oil section and a clean oil or suction section. The return line should discharge and return oil below the oil level in the tank to avoid trapping air. The oil in the return side rises and overflows the dividing wall into the clean oil side. Oil is drawn from the clean oil section of the tank by means of a floating suction rather than from the tank bottom where it may be contaminated with water, sludge and dirt.

Steam coils usually are provided in these tanks to heat the oil for starting operations in cold weather and to aid in separation of contaminants and moisture. The coils should be removable, but there should be no screwed joints inside the tank. Copper coils are not recommended because they corrode and leak very rapidly under certain conditions of moisture and when acids are developed by oxidation of the oil.

The reservoirs themselves should be adequately vented to provide for heavy condensation. They also should have water draw-offs in each section. When drawing water from the bottom of the tanks, the valve should be just cracked open; if it is opened too wide, oil may be drawn down through the water layer.

Two pumps are necessary components of the system, one acting in a reserve capacity. The pumps discharge into a common header line with check valves needed between each pump.

There should be a pressure gauge on the header line and a pump relief valve discharging back into the tank. An extra valve off the header line permits use of the system's pumps in pumping oil out of the tanks for periodic clean-outs.

FILTER MAINTENANCE

Oil leaves the header line through filters. These may be of the self-cleaning type, cylindrical cartridges made of a great variety of filtering material, manually cleaned screens or other devices. Filters containing Fuller's earth or similar material should not be used except with straight mineral oils, since they may remove the additives from oils.

Filters are protected by a relief valve, usually discharging to the pressure line. It is better to send some unfiltered oil to the equipment than no oil at all. An increased pressure differential in the filter indicates a dirty filter.

For cleaning purposes, the filter should have an air line connection in the top so that during shut-downs filter inlet and outlet valves may be closed and a valve opened at the bottom of the filter which drains oil back to the tank. The air inlet then may be opened and the filter charged with about 10 pounds air pressure. This pushes oil remaining in the filter back to the tank, and leaves the filter empty and clean for the resumption of operations.

The filter also should have a manual by-pass. In a continuous operating mill, it sometimes is necessary to by-pass the filter while repairs are made without shutting down the mill.

Most systems are equipped with oil coolers for without them it would be impossible to keep oil at a safe operating temperature. They should be of adequate size to avoid too much pressure loss and to provide sufficient cooling. Coolers should be installed so that the tube bundle can be readily removed for cleaning.

There is usually a pressure tank which serves several purposes. It acts as a surge tank, as a reserve

Nevertheless, difficulties are bound to arise. Here are some of the more common ones with suggestions as to how to cope with them:

tank and as an apparatus for operating protective pressure switches, signal lights, and stand-by pumps.

The pressure tank is partially filled to about one-half or two-thirds from the oil pressure line. Air pressure is fed into the top of the pressure tank to force the oil to the desired operating level. The air supply then is shut off. In the event of oil pressure failure, the oil level drops in the pressure tank and the air pressure in the tank decreases. At a predetermined setting, the first mercoid pressure switch closes and starts up the stand-by pump. This same switch may also sound an alarm to warn the operator that the second pump has cut in, indicating trouble. If the trouble (such as a broken pressure line) is not corrected, the pressure in the tank keeps dropping until the next mercoid switch cuts the control motors to the mill or stops the oil pumps.

A system such as this also should have a system pressure control valve which automatically regulates the pressure of oil going into the mill. This valve should be located in the pressure line after the filter and cooler.

An oil system requires regular attention and maintenance. It should be started about two hours before the mill is to operate, allowing time to detect and correct any difficulties without delaying the mill.

When the system is in operation it should be checked hourly; the oil lines to the various pieces of equipment, the sightoilers to bearings, the tel flows to sprays, pressures on the line, temperatures to bearings, gear cases, etc.

In checking the oil level in the reservoir, a quick glance at the mark previously placed on the level indicator will suffice. Oil temperature in the tank may be checked by the tank thermometer. Pressure is indicated by the pressure gauge on the feed line. Temperature of the oil going to the equipment is checked by the thermometer on the cooler outlet. The filter is checked by noting its differential pressure gauges. The oil level on the pressure tank is readily seen. With a touch of the hand to the oil pumps, the system is completely reviewed and trouble avoided.

If a good operating oiler is on the job, most troubles are avoided or corrected before serious damage is done. A well-engineered oil system is no better than the personnel who are responsible for maintaining it.

A. LOSS OF OIL

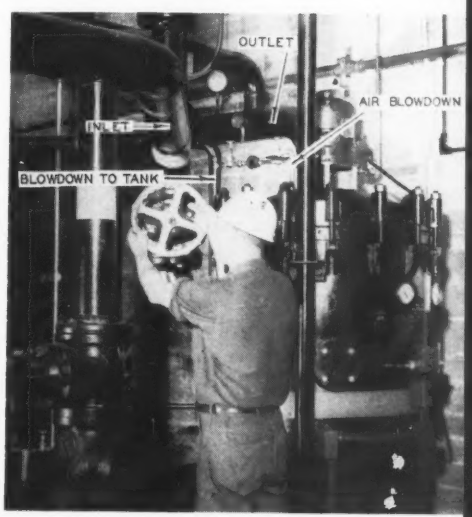
1. Remark oil level indicator on tank and note the time.
2. Check all lines and equipment for leaks.
3. Report estimated hourly oil loss to foreman. Remember: Oil loss may be cheaper than repair.
4. Determine if equipment is getting oil. Return line loss is not serious — only oil is lost. Feed line loss, however, may entail a risk to equipment.
5. Bypass and clean a dirty filter.

B. LOW OIL PRESSURE

1. Check oil level in reservoir — low oil level or stuck float suction may cause pumps to get air.
2. Check oil temperature—if thin from too much heat, oil can create pump slippage.
3. Check for a belt slip if pump is belt-driven.
4. Check relief valves. A pump check valve may be worn, allowing one pump to discharge through standby pump. Switch over to it.

C. WATER CONTAMINATION

Locate source and correct as soon as possible.



Pipe line set-up for an automatic filter unit.

☆

It's Toronto in '57

ASLE-ASME

4th Lubrication Conference

October 7-8-9 Royal York Hotel

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Lube Lines

by A. F. Brewer*

FIRE RESISTANCE

The corollary to fire resistance is safety. Safety is one of the cardinal precepts which must be observed when machinery is to be operated and lubricated; as such it must be considered from two viewpoints:

1. The human side, which could involve bodily injury should operators slip or fall on dripped or spilled lubricants.
2. The machinery side (which also may include injury to personnel) in case of fire caused specifically by flash or spontaneous ignition or combustion of flammable oils or oily materials.

The former can be prevented to a great extent by cleanliness and good housekeeping. The latter, while naturally of general interest to management as a whole, is of specific concern to those involved with operation of steam turbines, compressors, and the use of hydraulic fluids for power transmission. The fact that there have been flash fires has livened this interest, as indicated by the current activity in study of relative fire resistance.

Any petroleum product is flammable by nature, under conditions capable of supporting flash or combustion. Fortunately these conditions, while they naturally involve oxygen, also require a temperature at least as high as the flash point of the product, i.e., above 300° F where lubricants are concerned. In general production and industrial service a lubricant rarely should be directly subjected to such heat, except perhaps in oven conveyors and here a specialty lubricant of least flammability is used as a rule.

So we come back to the subject of safety as related to hydraulics, compressor and turbine lubrication. In direct-connected steam turbines an oil of comparatively low viscosity is used, i.e., around 32 centistokes (150 SSU) at 100°F. The flash point ranges normally above 400°F, the spontaneous ignition temperature being considerably higher. Since the generally accepted maximum operating temperature range in a turbine oiling system is from 150 to 180°F this would seem to afford an ample safety factor. However, under high temperature steam conditions which today are approaching 1200°F, the temperature of piping and turbine parts very close to the oil circulating lines and bearings can be so high (even with packing or covering) as to be blistering. Then again, some of this oil is under continual pressure, possibly as high as 50 to 200 psi and considerable gallonage is being circulated, i.e., 60 to 850 gallons per minute. This requires positive sealing of the bearings and a thoroughly tight oiling system throughout. Then the hazard normally is very slight. But seals can blow, or pipe fittings can fail. When this occurs a spray of oil onto hot turbine parts can result in a flash fire. J. J. O'Connor discussed this in his Progress Report on Fire Resistant Turbine Fluids before the ASLE annual meeting in 1956.

While synthetic materials of virtually negligible flammability are in the picture, it is too early to predict the extent to which they may supplant the accepted premium grade additive-type petroleum-base turbine oils in the foreseeable future. There are problems involving foaming, for example, which must be solved. Some difficulty in maintaining effective separation of solid contaminants also has been reported, especially where water in excess of 4 to 5 per cent prevails. Even so, the interest is widespread and considerable field testing may be expected to augment that which is already under way, with due consideration of the fact that the lubricating ability is a most important characteristic in any product under test. Meanwhile with existing systems and conventional oils, normal precautions to make sure that the lubricating system is designed and maintained to prevent possible oil leakage will afford a reasonable safety factor. A great number of successful operations are in service today. The responsibility of management is to see that they are *kept safe.*

Research incident to the development of fire-resistant compressor lubricants for air compressor lubrication is another note-worthy advancement in the interest of safety. This has resulted in the availability of wholly synthetic ester-type fluids, designed to minimize hazards of explosion and receiver fires and to reduce deposit formation on exhaust valves, in the interest of economical maintenance.

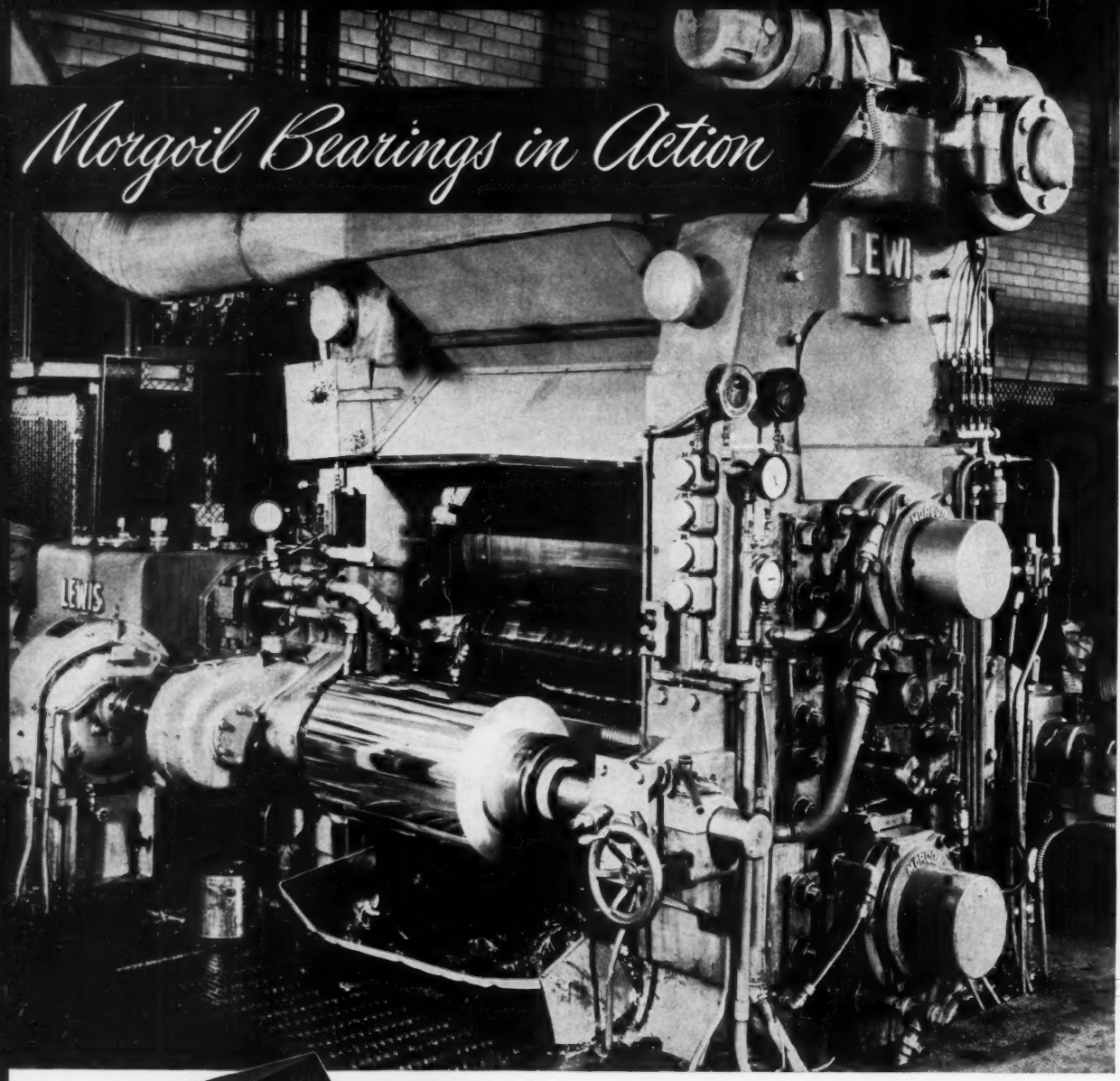
FIRE HAZARDS IN HYDRAULIC SERVICE

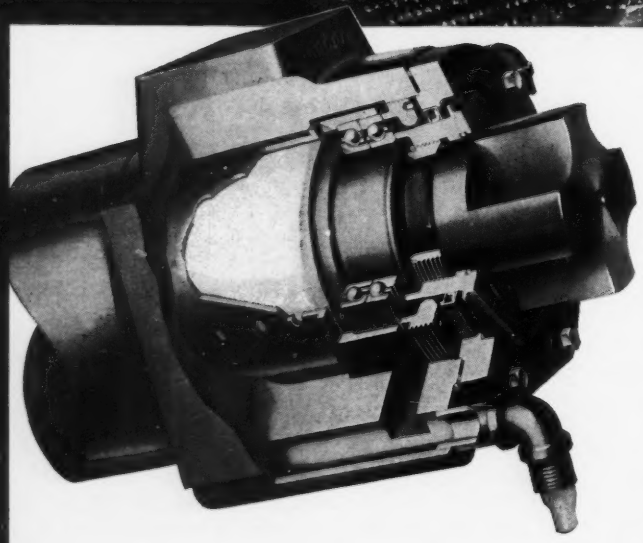
In hydraulic service the duty of the fluids involves lubrication only to a partial extent; the basic function is power transmission. On the other hand the wear preventive characteristic of the fluid must not be overlooked. Interest in fire resistance stemmed from the necessity in aircraft hydraulic systems to obtain a fluid which would be as safe and resistant to fire as possible. This led to the water-base and ester-type synthetic fluids which are accepted today for this purpose.

The question — "Why is there a fire hazard in hydraulic power transmission?" is pertinent. It comes back to the matter of temperature in the system and the flash propensity (by reason of the tendency towards spontaneous ignition) in the presence of air. The prevailing pressures in such systems may range as high as 2500 psi. A leak under such pressure could cause a spray of fluid virtually akin to that from a fire hose nozzle. Were this to flash or impinge upon a heated surface as might be present in die casting, furnace or steel mill operations, for example, a serious hazard could result. The Associated Factory Mutual Fire Insurance Companies have dramatically described how this could happen, in Factory Mutual Record for February 1957. Obviously relative fire resistance of the fluid must be considered. Cost must be evaluated with respect to insurance value, relation of production and safety of personnel. While it is a management responsibility to see that the most economical fluid is used in accordance with the service conditions, this fluid also *must be the safest*, in keeping with adequate lubricating ability.

*Consultant, and Author of *Basic Lubrication Practice.*

Morgoil Bearings in Action





The 18″-72″ MORGOILS on the 9″ and 21″ x 44″ Aluminum Foil Mill at Cochran Foil Co. pictured above make it easier to produce a quality product at high speed. Foil as thin as .00025 of an inch is rolled at speeds approaching 4000 feet per minute on MORGOIL-equipped mills. MORGOIL BEARINGS give you freedom from wear and fatigue and permit operation at high speeds with remarkable long bearing life.

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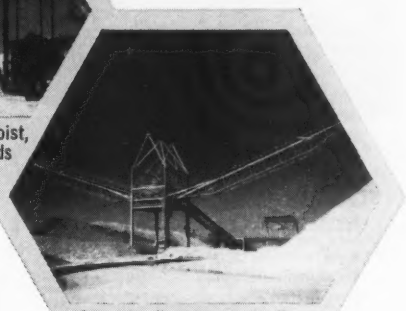
"In 14 months...not a single bearing lubrication failure with lithium-base grease!"

Conveyor-stacker handling moist, sticky material which builds up on the rollers.



Ore unloading conveyor rollers handling 200 tons of ore per hour.

THE PROOF IS IN THE PERFORMANCE...

lithium-base grease does the job...

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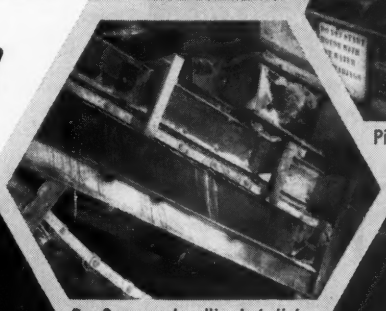
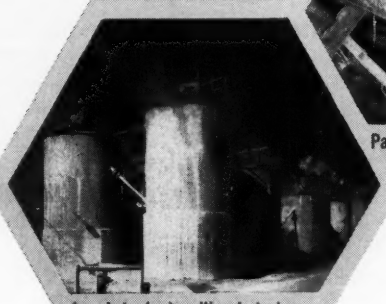
Pinion gear transmitting power from 600 h.p. motor to a ball mill.



Pan Conveyor handling hot clinker (1600°F), roller bearings in dusty, moist atmosphere.

V.A. Wemco classifier, part of wet grinding system, lower bearing submerged in hot sludge.



Leach tanks handling hot slurry, agitators driven by Falk gear reducing units.

Here's a report of our own experience with lithium-base grease under extreme industrial service conditions. Approximately 95% of the grease used in the plant of AMERICAN LITHIUM CHEMICALS, Inc., our subsidiary at San Antonio, Texas, is lithium-base, one-type grease. In fourteen months operation we have not been able to trace a single cause for bearing failure to the lubricant used. The on-the-spot photos above give graphic evidence of the rugged bearing service requirements in this plant where lithium ores are processed into high-grade lithium hydroxide, itself an important ingredient in lithium-base grease. Performance like this is why grease chemists, manufacturers, marketers and users all attest to the superiority of lithium-base... the *one* grease in place of *many* for efficient and economical operation.

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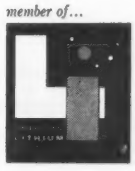

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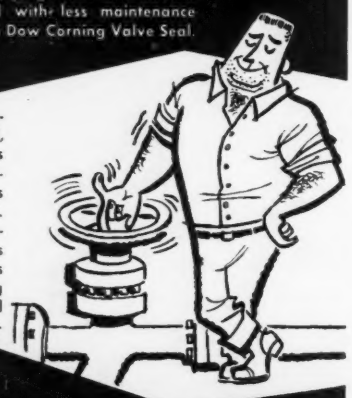

A nonmelting, grease-like silicone compound, Valve Seal retains its lubricating effectiveness at temperatures from —40° to 500°F. Resists moisture, oxidation, many chemicals and gases. Eliminates the need for stocking a variety of special purpose valve lubricants.

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Write for 8-page booklet,
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Summaries of Foreign Papers

"Metal Soaps (Stearates) for the Drawing of Steel Wire" by K. Schimz, *Draht*, Vol. 4, 1953, No. 8, p. 297.

The service life of wire drawing dies may be prolonged by using dry metal soaps, e.g., calcium stearate, as lubricants. Among their advantages are:

1. Low coefficient of friction at high pressures per unit area.
2. Corrosion protection of stock being drawn.
3. Easy removal of lubricant (no ignition residue).
4. Subsequent cold working of wire is facilitated, e.g., in the manufacture of bolts and screws.
5. Consumption of lubricant is low (0.10% of wire weight).

Aluminum and calcium stearates can be used for normal low-carbon steels; lead stearate for alloy steels. The stearates can be diluted with lime or with dry drawing soap. Table I gives information about various stearates.

TABLE I

Property	Type of Stearate		
	Aluminum	Calcium	Lead
Ignition Residue, %	10.7 as Al_2O_3	10.2 as CaO	31 as PbO
Melting Point, °C.	160	124	97
Coking Temp., °C.	.....	400-500	.....
Bulk Density, g/cc	0.102	0.125	0.163

(Abstractor's Note: Table I indicates ignition residues for the various stearates which is not entirely in agreement with the author's earlier statement of no ignition residues.)

Reviewed by E. H. Loeser

"Chemical Analysis of Surface Layers of Metals Exposed to Various Types of Wear," by B. I. Kostetskii, N. L. Golego, and P. K. Topekha, *Vestnik Mashinostroeniia USSR* 36 No. 10, pp. 25-26, 1956.

Under certain conditions of friction and wear (primarily the oxidation type of wear), a white, hard noncorrosive band has been observed within the surface layer of steels. Such bands accompany surface hardening which is caused by high momentary heating during friction. The formation of the bands was thought to be caused by oxygen enrichment of the surface layer, which was in turn associated with the formation of iron oxides during friction. The purpose of this article was to present data in support of this idea.

Chemical analysis of thin surface layers on ordinary wear specimens is notoriously difficult. The authors have developed a special procedure for obtaining specimens of surface layers for chemical analysis. This procedure consists of backing a steel foil with a steel block, and subjecting the foil to sliding conditions. After the test, the foil is removed and analyzed chemically. Wear tests were conducted in the KE-4 machine[1] under conditions of unlubricated sliding in air at a temperature of 20°C and with a humidity of 60%. The main analysis was the determination of oxygen content of the foil before and after four different types of wear. The results represent averages for 30 specimens subjected to each type of wear.

The four types of wear and the oxygen content of the metal foil for each type, as well as the oxygen content.

of the original metal, are shown in the following table:

Type of Wear[2]	Sliding Velocity, m/sec	Normal Pressure kg/cm²	O_2 Content, %
(Original metal)	....	....	0.0194
Seizure of Type I	0.25	15.0	0.0340
Oxidation	1.00	7.5	0.5184
Seizure of Type II	3.00	15.0	0.0661
Abrasive wear	0.25	7.5	0.0154

The amount of nitrogen in the foil was determined for all types of wear. Only slight traces were found.

The data show that the oxygen content of the foil after the oxidation type of wear is almost 30 times that of the original metal. The seizure type of wear gives an oxygen content two or three times greater than the original metal. The abrasive type of wear gives an oxygen content approximately the same as the original metal.

Thus, atmospheric oxygen plays an important part during the seizure types of wear and particularly during the oxidation type of wear. These results confirm the classification of the types of wear that had been developed by the authors. The authors believe that the experimental method used is distinguished by its simplicity and its accuracy, permitting them to obtain direct data on the chemical composition of surface layers.

REFERENCES

1. Kostetskii, B. I., "Wear Resistance of Machine Parts," Mashgiz, 1950.
2. Kostetskii, B. I., "Fundamental Problems of Wear Resistance of Machine Parts," Mashgiz, 1955.
3. Kostetskii, B. I., Eliseev, V. D., Preis, G. N., Methods and Machines for Wear Testing of Metals, Mashgiz, 1955.
4. Topekha, P. K., "Fundamental Types of Metal Wear," Mashgiz, 1952.

1. Not described in original article.
2. Not described further in original article.

Reviewed by Douglas Godfrey

"Influence of Lubricants upon the Surface Flow of Metal," S. Ya. Veiler, *Doklady Akademii Nauk USSR*, Vol. 99, 1954, No. 6, pp. 1025-7.

This study is concerned with the effect of lubricants on the flow in the surface layer of copper metal which is subjected to high pressures. A steel ball (6.316 mm. diam.) is slowly forced through a 6.00 mm. diameter hole in a copper bar. The required tangential force is measured and the microstructure and microhardness of the copper contact surface are investigated. Values listed in Table I show that lubricants lower the tangential force required to push the ball through the hole; water has little effect. As the steel ball is pushed through the hole, the surface layer of copper is plastically deformed and a bead of copper is formed ahead of the ball. This bead is larger under dry than under lubricated conditions. The required tangential force decreases on repeated passes of the ball since the copper metal is being continually squeezed toward the exit of the hole and after many passes the tangential force becomes constant. At this point, plastic flow ceases and the only force required is that needed to overcome elastic surface deformation. Table II indicates that the elastic recovery in the presence of an oxidized paraffin wax is greater than under dry conditions, and that more metal is plastically deformed under dry conditions. Microstructure studies were made

(Continued on p. 475)

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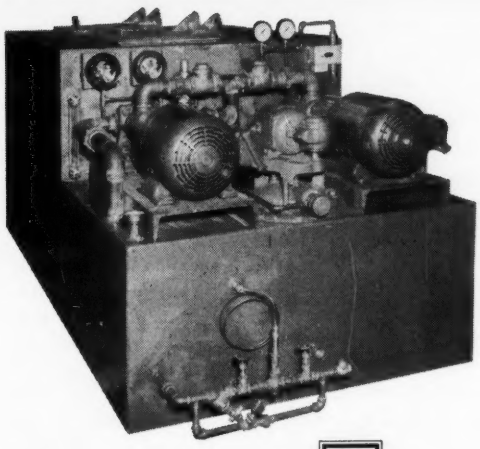

Designed especially for systems requiring up to 50 gpm, Dravo-DeLaval Unilube units contain the same engineering features found in the larger, custom designed systems. Every Dravo system includes four essentials of dependable and efficient mill lubrication:

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Piping, elbows and valves are carefully selected to hold pressure drops to a minimum and to provide non-turbulent flow.

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Since 3033-S is based on a relatively low-cost acid, pelargonic acid, long-range economics are favorable. Also, availability is not contingent on strategic, imported raw materials since pelargonic acid is made from abundantly available domestic fats and oils.

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Current Literature

ILLUSTRATED DISC FILTER DATA

— A new 16-page bulletin illustrating and describing Nugent Laminated Fiber Disc Filters has been issued by Wm. W. Nugent & Co., Inc.

The bulletin provides complete information on the design, construction, specifications and capacities of standard, high pressure and duplex models. Also included are special purpose assemblies, accessories and valuable installation data. A copy of Bulletin 7C may be secured by writing Wm. W. Nugent & Co., 3440 Cleveland St., Skokie, Ill.

REVISED DESCRIPTIVE BULLETIN —

Bulletin 106, revised to incorporate new operating information on the Model LFW-1 Lubricant-Friction-Wear testing machine, has been released by The Alpha Molykote Corporation, 65 Harvard Ave., Stamford, Conn.

Included in the new bulletin is a complete description of the machine's operation as well as a revised list of specifications in the English and Metric systems.

SPRAY LUBRICATION DISCUSSED

— A technical paper, "Development and Application of Spray Lubrication," by E. J. Gesdorf, senior application engineer, is now available. The illustrated treatise combines application photographs, cutaway views of equipment and engineering drawing reproductions with a discussion covering the historical development of spray lubrication, operating details, nozzle theory, spray pattern evaluation and case studies of savings realized by users. Copies of this paper can be obtained from The Farval Corporation, 3267 E. 80th St., Cleveland 4, Ohio.

PUMP AND DRIVING MOTOR UNITS

A new series of positive displacement pump-and-driving-motor units has been announced for singular plant applications not warranting custom design. The units can be used in varying combinations for hydraulic, oil burning, lubricating and other services involving many different fluids.

The pump and driving motor units are close-coupled with the pump mounted on the motor thus eliminating the need for coupling, adapter or base. They reportedly provide space savings, reduced weight, fewer components, and less assembly time. For further information, write to Tuthill Pump Co., 939 E. 95th St., Chicago 19, Ill.

TURBINE FLOW METER CATALOG

— Fischer & Porter Co., manufacturer of complete process instrumentation, announces the publication of a catalog on the F&P Turbine Flow Meter, a liquid flow measurement device. Inserted directly into process piping, the meter measures flow rate of a variety of liquids having flows of from less than one gpm to more than 2500 gpm. Accuracy of the meter is $\pm\frac{1}{2}\%$ of instantaneous rate under production conditions.

The meter is designed so that mounting in any position has no effect on flow coefficient. Special thrust plates, installed in both front and rear hub assemblies, guard against effects of reverse flow. Catalog copies are available from Fischer & Porter Co., 75 Jacksonville Rd., Hatboro, Pa.

SURFACE ROUGHNESS DEFINED

— A 24-page American Standard booklet on Surface Roughness, Waviness and Lay, recently published by the American Society of Mechanical Engineers, is now available.

Compiled by the ASA-authorized sectional committee on classification and designation of surface qualities, the booklet shows manufacturers how to reduce machining costs and control surface finish. Geometric irregularities of solid material surfaces and physical specimens for gauging roughness are defined. Classifications are established for roughness, waviness and lay, and a set of symbols for drawings, specifications, and reports is included. To assure a uniform measurement basis, specifications also are provided for precision reference specimens, roughness comparison specimens and tracer type instruments. Copies of the Standard may be obtained from Brush Electronics Co., 3405 Perkins Ave., Cleveland 14, Ohio.

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Synthetic Ester Lubricants

R. S. Barnes, Research Department, Standard Oil Co. (Indiana), Whiting, Indiana
M. Z. Fainman, Inland Testing Laboratories, Cook Electric Co., Morton Grove, Ill.

Synthetic esters have properties that make them good lubricants under conditions too severe for mineral oils. Raw materials for making them are available in quantity from petroleum sources. Early research in Germany led to commercial production, a good part from polyhydric alcohols and monobasic acids. Development in Britain and the U.S.A. concentrated on esters derived from monohydric alcohols and dibasic acids.
Lubricants with particularly good properties can be made from polyhydric alcohols and monobasic acids. Alcohols derived from neopentane have no hydrogen atoms on the *beta* carbon atom and give esters of improved thermal stability. Straight-chain monobasic acids give good lubricity. Such esters improve the lubricity of silicone oils as well as mineral oils.

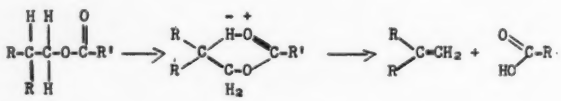
INTRODUCTION

Inability of mineral oils to lubricate over a wide temperature range has led to synthesis of lubricants for military, industrial, and automotive applications. Many different organic chemicals and polymers have been studied and used[1–7]. The excellent properties of certain esters have made them particularly attractive as lubricant bases or as materials to upgrade mineral oils [8–11]. Although these esters lubricate present aircraft gas turbines, they are inadequate for future requirements. The turbo-prop engine requires oils of good lubricity because of high loads. The turbo-jet engine is expected to operate at temperatures as high as 500°F. If a single lubricant is to serve both types of aircraft, it must have better lubricity and thermal stability than present ester lubricants.

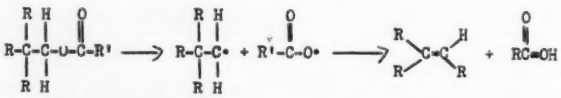
Synthetic esters were developed in Germany before and during World War II[9–11] as lubricants for low-temperature service. More than 3000 compounds were prepared and studied systematically. About 100 barrels per day of esters were used to upgrade mineral oil or other synthetic oils. These esters were made from Leuna alcohols and dibasic acids, and from polyhydric alcohols and Leuna acids.[10] Development in Great Britain and the United States was brought about by the severe requirements of the aircraft gas turbine and emphasized diesters derived from dibasic acids and primary alcohols, such as 2-ethylhexanol and Oxo alcohols derived from petroleum.

Structure has been reported to have little effect on the lubricity of esters except for slight improvement with increasing molecular weight[9–11, 19]. The Bowden stick-slip machine and the Boerlage 4-ball machine indicate that esters resemble mineral oils in friction and wear characteristics[18].

However, structure strongly influences the thermal stability of esters. They are more stable if they contain no hydrogen on the *beta* carbon atom of the alcohol portion[12]. When such hydrogen is present, esters decompose to form acids and 1-alkenes, presumably by a cyclic mechanism:



When *beta* hydrogen is absent, cyclization cannot take place. Decomposition requires free-radical mechanism:

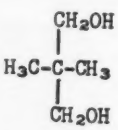


Because free radicals require more energy to form, a higher initiating temperature is required for decomposition. Thus, esters of dibasic acids and tertiary-alkyl carbinols are more stable than those made from the isomeric normal-alkyl carbinols[13, 14].

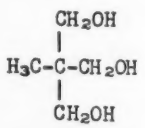
The problem is to obtain esters in which the alcohol portion has no *beta* hydrogen. Suitable primary monohydric alcohols for reaction with dibasic acids are not commercial materials. However, polyhydric alcohols derived from neopentane are available. Forty-eight esters made from such alcohols and monobasic acids were synthesized and compared in lubricity and thermal stability with six dibasic-acid esters.

SYNTHESIS OF ESTERS

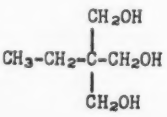
Six polyhydric alcohols derived from neopentane were obtained from commercial sources. Four were industrial materials:



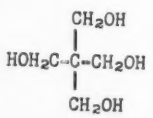
Neopentyl glycol



Trimethylolethane

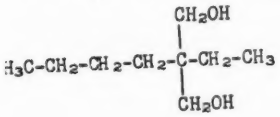


Trimethylolpropane

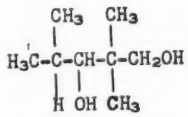


Pentaerythritol

Two were exploratory chemicals:



2-butyl-2-ethyl-1,3-propanediol



2,2,4-trimethyl-1,3-pentanediol

Acids were obtained from three sources. Normal acids were commercially available products derived from natural sources. Highly branched acids were made by oxoation of four types of petroleum olefins: mixed butylene-propylene dimer, butylene dimer, propylene trimer, and butylene trimer. Mixed normal and slightly branched acids were obtained by caustic extraction of the liquid product from hydrogenation of carbon monoxide[15].

The esters were made by refluxing a solution of the polyhydric alcohol and five mol percent excess acid in toluene in the presence of benzene-sulfonic acid as a catalyst. Water formed was removed continuously by azeotropic distillation with toluene through a Stark and Dean receiver. The final reaction mixture was washed thoroughly with dilute caustic and dried. Excess toluene was removed and the ester was then distilled at a pressure of about 0.1 mm.

The effect of structure on physical properties follows the principles outlined by previous investigators[5, 8]. Physical properties of the 48 esters are given in Table I. Esters from normal acids have high viscosity index, flash point, and pour point. Those from branched acids have lower viscosity index, flash point, and pour point. However, esters from mixed acids have high flash point and viscosity index but low pour point. The pour points of the esters from normal acids and trimethylolpropane are lower than those from neopentyl glycol, trimethylolethane,

TABLE I
Properties of Polyhydric-Alcohol Esters

Acid	Density 20 d_4	Index of Refraction n_D20	Pour Point, °F	Flash Point, °F	Viscosity, Cs. 100°F	Viscosity, Cs. 210°F	Viscosity Index	ASTM Slope
			Neopentyl Glycol					
n-C_9	0.9160	1.4447	− 5	410	9.12	2.62	136	0.72
n-C_{10}	0.9091	1.4467	− 5	435	11.3	3.01	141	0.72
n-C_{14}	—	—	93*a*	—	—	—	—	—
br-C_8	0.9238	1.4436	< −65	385	9.22	2.43	91	0.79
br-C_9	0.9110	1.4430	< −65	380	14.0	3.07	78	0.79
br-C_{10}	0.9148	1.4485	< −65	395	15.2	3.19	73	0.80
br-C_{13}	0.9110	1.4548	−65	460	35.0	5.28	88	0.77
			Trimethylolethane					
n-C_9	0.9425	1.4522	+35	535	21.0	4.48	147	0.70
n-C_{10}	0.9329	1.4536	+ 5	535	24.5	5.12	150	0.68
n-C_{14}	—	—	115*a*	—	—	—	—	—
br-C_8	0.9553	1.4520	−85	480	27.1	4.72	103	0.76
br-C_9	0.9416	1.4510	−55	465	54.9	6.85	85	0.77
br-C_{10}	0.9425	1.4558	−55	510	46.5	6.42	95	0.75
br-C_{13}	0.9315	1.4608	−45	545	111.0	10.41	80	0.74
mix-C_5	0.9913	1.4428	< −90	425	10.1	2.57	90	0.80
mix-C_6	0.9757	1.4451	< −80	425	12.2	2.91	96	0.78
mix-C_7	0.9632	1.4488	< −80	465	15.3	3.42	110	0.75
mix-C_8	0.9548	1.4511	< −80	485	19.1	3.92	113	0.75
			Trimethylolpropane					
n-C_6	0.9726	1.4417	< −80	470	12.5	3.06	115	0.75
n-C_7	0.9608	1.4505	< −80	495	15.2	3.51	125	0.74
n-C_8	0.9509	1.4525	−65	505	19.0	4.09	135	0.72
n-C_9	0.9447	1.4545	−55	525	22.9	4.62	135	0.71
n-C_{10}	0.9352	1.4553	−20	540	27.6	5.42	142	0.69
n-C_{14}	—	—	107*a*	—	—	—	—	—
br-C_8	0.9578	1.4543	−65	460	31.9	5.07	92	0.78
br-C_9	0.9409	1.4528	−40	445	59.7	7.30	88	0.75
br-C_{10}	0.9407	1.4573	−60	530	50.8	6.80	95	0.75
br-C_{13}	0.9307	1.4617	−35	535	117.5	10.94	82	0.74
mix-C_4	1.0152	1.4439	< −90	390	9.59	2.42	74	0.81
mix-C_5	0.9906	1.4456	< −90	425	11.5	2.77	88	0.79
mix-C_6	0.9744	1.4479	< −80	435	13.9	3.13	92	0.78
mix-C_7	0.9629	1.4507	< −90	470	17.2	3.63	105	0.76
mix-C_8	0.9554	1.4530	< −80	465	21.7	4.26	115	0.76
mix-C_{9-10}	0.9473	1.4560	−80	495	27.3	4.89	115	0.73
			Pentaerythritol					
n-C_9	0.9577	1.4567	+30	555	34.7	6.23	135	0.68
br-C_8	0.9729	1.4567	−60	495	55.6	7.39	102	0.73
br-C_9	0.9552	1.4550	−30	505	129.2	11.60	82	0.73
br-C_{10}	0.9554	1.4598	−45	520	92.1	9.85	93	0.74
mix-C_5	1.0212	1.4485	< −90	475	20.6	3.98	97	0.77
mix-C_6	1.0006	1.4506	−80	495	24.5	4.50	107	0.75
mix-C_7	0.9821	1.4531	< −90	500	26.8	4.97	124	0.72
mix-C_8	0.9763	1.4557	−70	530	37.8	5.94	111	0.73
mix-C_{9-10}	0.9621	1.4582	−75	535	47.3	7.07	116	0.70
			2-Butyl-2-ethyl-1,3-propanediol					
n-C_9	0.9135	1.4512	< −80	450	14.4	3.34	117	0.69
n-C_{14}	0.8975	1.4568	+30	540	30.2	5.92	146	0.67
n-C_{18}	—	—	+70	575	45.0	8.06	142	0.62
			2,2,4-Trimethyl-1,3-pentanediol					
n-C_9	0.9132	1.4487	< −80	420	12.0	3.08	133	0.69
n-C_{10}	0.9068	1.4500	−45	460	14.4	3.52	144	0.72

a. Melting point, °F.

and pentaerythritol. The greater branching of 2-butyl-2-ethyl-1,3-propanediol and 2,2,4-trimethyl-1,3-pentanediol leads to still lower pour points. Three esters proved to be solids and were not further studied.

TABLE II
Boundary Lubrication Properties of Polyhydric-Alcohol Esters

Acid	Co-efficient of Friction Static, μ_s	Kinetic, μ_k	μ_s/μ_k
	Neopentylglycol		
n-C_9	0.131	0.142	0.92
n-C_{10}	0.120	0.137	0.88
br-C_8	0.183	0.181	1.01
br-C_9	0.213	0.192	1.11
br-C_{10}	0.188	0.184	1.02
br-C_{13}	0.187	0.184	1.02
	Trimethylolethane		
n-C_9	0.109	0.129	0.79
n-C_{10}	0.104	0.126	0.83
br-C_8	0.169	0.167	1.01
br-C_9	0.224	0.197	1.14
br-C_{10}	0.184	0.181	1.02
br-C_{13}	0.184	0.181	1.02
mix-C_5	0.178	0.175	1.02
mix-C_6	0.179	0.188	0.95
mix-C_7	0.156	0.167	0.93
mix-C_8	0.137	0.157	0.87
	Trimethylolpropane		
n-C_6	0.112	0.135	0.83
n-C_7	0.126	0.142	0.89
n-C_8	0.113	0.131	0.86
n-C_9	0.121	0.142	0.85
n-C_{10}	0.092	0.118	0.70
br-C_8	0.186	0.183	1.02
br-C_9	0.205	0.181	1.13
br-C_{10}	0.181	0.177	1.02
br-C_{13}	0.183	0.181	1.01
mix-C_4	0.180	0.177	1.02
mix-C_5	0.184	0.182	1.01
mix-C_6	0.160	0.174	0.92
mix-C_7	0.157	0.171	0.92
mix-C_8	0.166	0.181	0.92
mix-C_{9-10}	0.147	0.167	0.88
	Pentaerythritol		
n-C_9	0.089	0.122	0.73
br-C_8	0.171	0.169	1.01
br-C_9	0.190	0.176	1.08
br-C_{10}	0.174	0.172	1.01
mix-C_5	0.203	0.201	1.01
mix-C_6	0.158	0.172	0.92
mix-C_7	0.134	0.158	0.85
mix-C_8	0.143	0.165	0.87
mix-C_{9-10}	0.136	0.157	0.87
	2-Butyl-2-ethyl-1,3-propanediol		
n-C_9	0.112	0.129	0.87
n-C_{14}	0.120	0.136	0.88
n-C_{18}	0.087	0.104	0.84
	2,2,4-Trimethyl-1,3-pentanediol		
n-C_9	0.148	0.157	0.94
n-C_{10}	0.119	0.135	0.88

LUBRICITY

The lubricities of the 45 liquid polyhydric alcohol esters, as well as those of six dibasic acid esters, were determined under boundary conditions. The values of the static (μ_s) and kinetic (μ_k) coefficients of friction were obtained on a stick-slip machine[17]. Although the mathematical analysis of the stick-slip process is based on a simple approximation, the friction coefficients obtained permit comparison of the lubricants. Smooth sliding occurs when $\mu_s/\mu_k = < 1$.

As shown in Table II, the esters of normal acids and polyhydric alcohols are good boundary lubricants. Lubricity varies with the structure of the alcohol, as is readily seen by comparing the esters of n-nonanoic acid. Although all give smooth sliding, the coefficient of friction varies more widely than would be expected from differences in viscosity. Those made from highly branched acids are poor lubricants; those made from mixed acids are intermediate in lubricity between esters derived from normal and highly branched acids.

As shown in Table III, esters made from alcohols and dibasic acids are poor boundary lubricants regardless of the structure of the alcohol.

The lubricity of polyesters derived from normal acids suggested using them as "oiliness" agents in other fluids. Table IV shows they markedly lower the coefficient of friction of a mineral oil and a silicone oil, but are only moderately effective in another ester.

TABLE III
Boundary-Lubrication Properties of Dibasic-Acid Esters

Alcohol	Acid	Co-efficient of Friction Static, μ_s	Kinetic, μ_k	μ_s/μ_k
br-Octanol	Adipic	0.262	0.174	1.51
br-Octanol	Azelaic	0.190	0.173	1.10
2-Ethylhexanol	Sebacic	0.189	0.173	1.12
n-Octanol	Adipic	0.241	0.153	1.58
n-Decanol	Adipic	0.200	0.164	1.22
br-Tridecanol	Adipic	0.222	0.183	1.22

THERMAL STABILITY

Other investigators have determined thermal stability in an inert atmosphere by measuring weight loss, change in viscosity, and change in pressure[12–14, 16]. However, below 575°F, esters give acids in addition to other products of decomposition[12, 20]. Therefore, the ratio of the neutralization number to the saponification number was used to measure the extent of decomposition. Selected

TABLE IV
Boundary-Lubrication Properties of Polyhydric-Alcohol Esters in Blends

	Coefficient of Friction Static, μ_s	Kinetic, μ_k	μ_s/μ_k
Mineral oil, SAE 10 grade	0.220	0.178	1.24
Mineral oil + 10% neopentyl-di-n-nonoate	0.153	0.151	1.01
Mineral oil + 20% neopentyl-di-n-nonoate	0.151	0.160	0.94
Silicone oil	0.299	0.294	1.02
Silicone oil + 10% trimethylolpropane-tri-n-nonoate	0.153	0.170	0.90
Silicone oil + 20% trimethylolpropane-tri-n-nonoate	0.141	0.160	0.87
Silicone oil + 10% neopentyl-di-n-nonoate	0.160	0.174	0.92
Silicone oil + 20% neopentyl-di-n-nonoate	0.150	0.161	0.93
20% neopentyl-di-n-nonoate + 80% trimethylolpropane-tri-br-nonoate	0.186	0.183	1.02
50% neopentyl-di-n-nonoate + 50% trimethylolpropane-tri-br-nonoate	0.145	0.167	0.87

esters were decomposed in evacuated quartz tubes maintained at 500°F in an aluminum block.

The rates of decomposition are compared in Figure 1. Esters from trimethylolpropane and pentaerythritol have no *beta* hydrogen and are the most stable at 500°F. Di-br-octyl azelate which has *beta* hydrogen decomposes almost three times as fast.

2,2,4-Trimethyl-1,3-pentane-di-n-nonoate and 2-butyl-2-ethyl-1,3-propane-di-n-nonoate do not behave as expected. Even though it contains one *beta* hydrogen the former is the most stable. However, both decompose so rapidly that esters with long alkyl chains in the *beta* position apparently react in a manner different from either the cyclic or the free radical mechanism. Further study will be needed to resolve this anomaly.

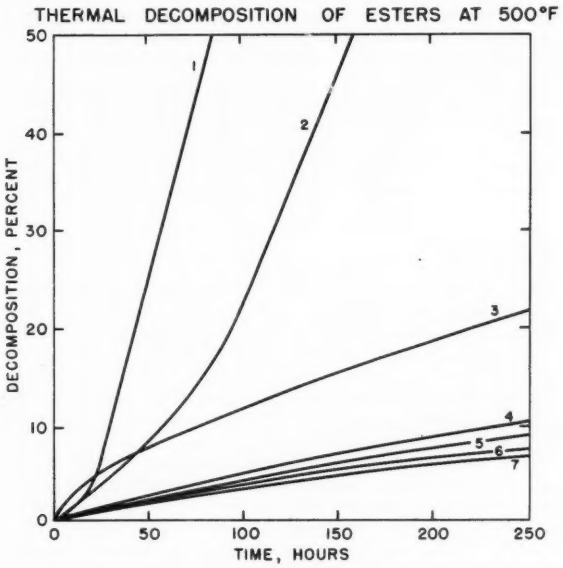


FIGURE 1.

1. 2-BUTYL-2-ETHYL-1,3-PROPANE-DI-n-NONOATE
2. 2,2,4-TRIMETHYL-1,3-PENTANE-DI-n-NONOATE
3. DI-br-OCTYLAZELATE
4. TRIMETHYLOLPROPANE - TRI - br - OCTANOATE
5. TRIMETHYLOLPROPANE - TRI - mix - OCTANOATE
6. TRIMETHYLOLPROPANE - TRI - n - NONOATE
7. PENTAERYTHRITOL - TETRA - n - NONOATE

CONCLUSION

Esters can be synthesized from polyhydric alcohols to give both good lubricity and good thermal stability in a single molecule. Such materials improve the lubricity of silicone oil as well as mineral oil. Production of the raw materials from petroleum sources should make large volumes of such esters available at low prices. These esters look promising for use in present and future machines, but considerable development work remains to be done.

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PREPARED DISCUSSION OF THE PAPER, "SYNTHETIC ESTER LUBRICANTS WITH IMPROVED LUBRICITY AND THERMAL STABILITY"

by *W. A. Zisman and H. Ravner*

Fainman and Barnes have chosen one of the sound approaches to the problem of developing more heat-stable esters. Several other laboratories have also improved the stability of esters by eliminating the hydrogen on the beta carbon atom. However, one limitation of the resulting compounds as lubricants is that this synthetic change leads to excessive branching which has an adverse effect on the viscosity-temperature relationship, and this may greatly limit their lubricant application. This is not to imply that other classes of compounds which are more heat stable than conventional diesters, are any better in this respect than the products reported here. One of the difficulties in synthesizing high V. I. liquids is that modification of the molecular structure to improve one property usually results in adverse effects on other desirable properties.

Although the authors do not specify which viscosity index they employ, simple calculations indicate that they have used the viscosity index of Dean and Davis. It is well known that this index is so dependent on the viscosity that it is difficult to use it to arrive at valid comparisons between oils of different molecular structures. The Hardman and Nissan modification of the viscosity index is a more valid measure for such uses since it is much less dependent on the viscosity.

Several questions come to mind regarding the authors' presentation of their friction data. Such essential information as the rubbing metals used, the test temperatures, the relative slide speeds, and the load (all of which may greatly effect the results) are not even mentioned. When one makes the reasonable assumption that their data relate to steel sliding on steel at slow speeds and at ordinary temperatures, the results obtained are still puzzling. The differences reported by the authors in the static coefficients of friction of diesters made from branched and straight-chain acids appear to us to be much greater than would be expected with pure diesters. By "pure" we refer to compounds from which polar impurities (especially those of high molecular weight) have been removed by suitable absorbents shortly before the friction measurements are made.

The authors do not indicate that such essential procedures were employed. The presence of only trace quantities of a free acid such as unbranched C_9 could reasonably be expected to lower the coefficient of static friction at room temperature to values from 0.10 to 0.13 because of the close-packed monolayer adsorbed on the metals. Equimolal concentrations of branched acids could not form such close-packed monolayers because of steric hindrance from the side chains. Consequently, the coefficient of the friction would not decrease to values as low as 0.10 to 0.13. Although the differences in the lubricity of esters made from straight-chain and from branched-chain acids would be expected to be in the same direction as that reported by the authors, they should not be so large. We have shown in previous publications that a variety of types of pure esters have coefficients of friction for steel on steel at 20°C between 0.20 and 0.26. The data presented would be far more valuable if they included the static coefficient of friction of each compound after proper purification. In reporting their results on esters of lesser purity the authors have sacrificed their right to relate molecular structure to relative lubricity unless they can prove that equimolal concentrations of these acids are present in the esters compared.

One great advantage resulting from the use of esters as lubricants is that in their operation in most machinery, the products of oxidation and hydrolysis are usually long-chain fatty acids which are excellent wear- and rust-preventive addition agents. The longer the chain length of the acids and the less the branching, the greater will be the resulting improvement in the boundary lubricating properties. Because the lubricity data presented by the authors probably has been effected by the presence of such acids as impurities in their materials, they have credited their esters with better lubricating properties than others reported in the past. We believe their best esters have lubricity properties typical of many of the esters now in common use when they are used in an equally impure condition.

The data of Table IV indicate that some of the esters are able to lower the coefficient of friction of a silicone fluid to values as low as 0.14. Their data are inadequate to prove that the boundary lubricating difficulties with silicones can be cured in practice by the addition of these esters. The improvement reported was in all probability caused by the free acid present in the esters. Presumably the esters also act as a mutual solvent for the silicone and the acid. A study of the effect of higher speeds of sliding (especially under dynamic conditions, and of the effect of elevating the temperature considerably will be required to establish that their results represent a useful accomplishment.

AUTHORS' COMMENTS ON THE PREPARED DISCUSSION

1. We make no claim of originality in improving the stability of esters by eliminating the hydrogen on the beta carbon atom by citing the work of Sommers and Crowell and that of Blake, et al. If the discussors know of some additional pertinent references not mentioned in these papers, we will be glad to include them. We do feel that the use of commercially available alcohols and acids to produce esters with such desirable properties is worthy of mention.

2. On the question of viscosity index, viscosity temperature data on the esters discussed in our paper are presented in two ways—as Dean and Davis Viscosity Index and as ASTM slope. This was done because we feel that the viscosity at one temperature and the ASTM slope is the best way of comparing the viscosity-temperature behavior of similar fluids which do not vary too widely in viscosity. While no viscosity index system enjoys universal acceptance, the Dean and Davis Viscosity Index values were included because it is the system most widely used in the lubrication field and is the ASTM Method.

3. It is true that changes in structure often adversely affect viscosity-temperature properties. However, we doubt that this effect is great enough in the case of the esters we reported to "greatly limit their application." A comparison of the viscosity-temperature properties of diesters of 2-Ethylhexanol and polyesters of trimethylol propane and normal acids points up their similarity.

VISCOSITY-TEMPERATURE BEHAVIOR OF ESTERS

Alcohol	Acid	Viscosity, CS. 100°F	210°F	ASTM Slope*	Viscosity Index Dean Davis	Hardiman Nissan
Trimethylol-	n-C_6	12.5	3.06	0.75	115	136
propane	n-C_7	15.2	3.51	0.74	125	132
	n-C_8	19.0	4.09	0.72	135	130
	n-C_9	22.9	4.62	0.71	135	128
	n-C_{10}	27.6	5.42	0.69	142	132
2-Ethyl-	Adipic†	8.22	2.38	0.77	120	148
hexanol	Pimelic†	9.28	2.65	0.74	137	151
	Azelaic†	11.4	3.06	0.72	146	148
	Sebacic†	12.6	3.32	0.71	154	149

*Range 100°F to 210°F
†From reference 8 cited in paper
‡From reference 6 cited in paper

4. It is not necessary to make any assumptions regarding the equipment or test conditions used to obtain our friction data. All of this information is given in the Merchant article referred to in our paper. However, it may be that this article was not available to the discussors, in which case they are repeated—Meehanite cast iron blocks moving at a relative speed of ¾ inch per minute under a load of 15 pounds. Temperature of test is 23 to 25°C.

The discussors have quite rightly raised the question of the purity of our esters. Because of the method of preparation of these esters, they were hydroxyl-free and contain less than 0.05% free acid. In many instances the acid content was less than 0.001%. When n-nonanoic acid is added to diisooctyl adipate, the concentration must be greater than 0.11% before smooth sliding occurs. Where normal acids were used to prepare the esters, the products have since been further purified by the method of Fox and Zisman.* The results reported in our paper were reproduced with these further purified samples.

Zisman and Ravner have pointed out the advantage of using esters of the type we have described; that their decomposition gives straight chain fatty acids which improve the boundary lubricating properties. However, it is not due to impurities of these acids as they say, but rather due to the reaction of these esters with iron oxide and water at the rubbing surface. This phenomenon was first pointed out by Bowden and Tabor† for ethyl stearate. Because dibasic acid esters and polyesters of branch chain acids cannot decompose to give acids which will form close-packed monolayers, they would not be expected to show good boundary lubricating properties. In our work such was the case. It should also be pointed out that the friction values we reported for diesters are in good agreement with those values obtained by Brophy and Zisman for similar materials.

*J. Coll. Science5, December 1950, p. 516
†"The Friction and Lubrication of Solids," Oxford University Press, London, 1950, p. 215

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Numerical Solution of Reynolds' Equation for Sector Thrust Bearings

A numerical technique employing a digital computer has been investigated to determine whether a solution of the Reynolds' equation could be obtained which would accurately predict sector bearing performance. A method was discovered which greatly increased the efficiency of the computations while maintaining a high degree of accuracy. The results compare favorably with experiment and known analytical solutions. Agreement with experiment was improved when an experimentally determined viscosity distribution was used. The investigation indicates that good design criteria may be established for sector bearings with arbitrary geometries; and possibly very accurate predictions may be obtained if the Reynolds and energy equations are solved simultaneously by this technique.

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SYMBOLS

Bar above symbols indicates per unit

		UNITS
U	— Velocity	in./sec
N	— rps	rps
Q	— Flow	gpm
R	— Outer radius of pad	in.
r	— Radius	in.
r_i	— Dimensionless radius	
L	— Radial length of pad	in.
θ_o	— Angle subtending the pad	radians
θ_i	— Dimensionless angle	
P	— Pressure	psi
$P_{k'}$	— Dimensionless pressure after final iteration	
F_i	— Dimensionless vertical force	
q	— Dimensionless flow	
μ	— Viscosity	lb-sec/in.^{2}
μ_{av}	— Average viscosity	lb-sec/in.^{2}
h	— Film thickness	in.
h_1	— Minimum film thickness	in.
b	— Taper	in.
k	— Number of iterations	

INTRODUCTION

This paper presents a numerical method employed for the solution of Reynolds' equation for sector thrust bearings. Using this technique, it is possible to arrive analytically at the optimum film shape and bearing geometry that will give the largest load-carrying capacity per prescribed minimum film thickness. The paper also compares solutions obtained on the 650 IBM digital computer with those obtained by Michell and Saibel.

In optimizing the thrust bearing design, it is necessary to calculate the load-carrying capacity as a function of film shape. Because of the complexity and length of the calculations, it was decided to program these calculations on an automatic digital computer, namely the IBM 650. In order to obtain general solutions, the Reynolds' equation was set up in dimensionless form. Using numerical methods, this partial differential equation was further transformed to a difference equation and, by the process of iteration, the pressure distribution in the field was determined. Trapezoidal integration was employed in calculating the load-carrying capacity.

The computer was also programmed to calculate the hydrodynamic oil flow across each edge of the pad and for the location of the pivot. This location of the pivot satisfies the equilibrium of moments in both the radial and circumferential directions. The paper discusses computations which were made for two film functions, Michell and tapered land. Results were compared with Michell's exact solution and Saibel's solution. Using an experimentally determined temperature distribution on the tapered land bearing, calculations were made to compare the effects of variable viscosity.

NUMERICAL SOLUTION OF THE REYNOLDS' EQUATION FOR SECTOR THRUST BEARINGS USING IBM COMPUTER

Pressure Calculation

The thrust bearing pad (shoe) is divided into (m x n) mesh as shown in Figure 1. There are m divisions in the θ direction and n divisions in the r direction. θ_0 is the angle subtended by the bearing pad. The pressure is evaluated at specific points in the mesh, and then the resultant force is determined by integration of the pressure over the area.

Presented at the ASLE Annual Meeting, April 5, 1956.

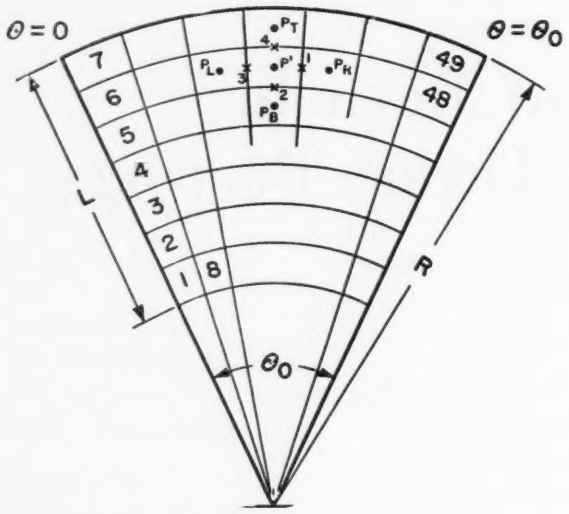

Fig. 1. 7 x 7 Thrust Pad Mesh

The basic Reynolds' equation in dimensionless form is:

$$\frac{\partial}{\partial \bar{r}}\left(\frac{\bar{r}\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{P}}{\partial \bar{r}}\right) + \frac{1}{\bar{r}}\frac{\partial}{\partial \bar{\theta}}\left(\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial \bar{P}}{\partial \bar{\theta}}\right) = 12\pi\bar{r}\left(\frac{R}{L}\right)^2\frac{\partial \bar{h}}{\partial \bar{\theta}} \qquad [1]$$

where $r = \bar{r}R$

$\theta = \bar{\theta}$

$U = 2\pi\bar{r}RN$

$\mu = \bar{\mu}\mu_{av}$

$h = \bar{h}h_1$

$P = P\mu_{av}N\left(\frac{L}{h_1}\right)^2$

From this point on, the bars above the symbols will be omitted for simplicity. Evaluating the derivatives by means of differences, Equation (1) is rewritten to:

Refer to Figure 1.

$$\frac{\dfrac{r_4h_4^3}{\mu_4}\left(\dfrac{P_T - P'}{\Delta r}\right) - \dfrac{r_2h_2^3}{\mu_2}\left(\dfrac{P' - P_B}{\Delta r}\right)}{\Delta r} + \frac{\dfrac{1}{r_i}\left[\dfrac{h_1^3}{\mu_i}\left(\dfrac{P_R - P'}{\Delta\theta}\right) - \dfrac{h_3^3}{\mu_3}\left(\dfrac{P' - P_L}{\Delta\theta}\right)\right]}{\Delta\theta} = 12\pi_{ri}\left(\frac{R}{L}\right)^2\left(\frac{h_1 - h_3}{\Delta\theta}\right) \qquad [2]$$

Solving for $P' =$

$$\frac{\dfrac{r_4h_4^3}{\mu_4\Delta r^2}P_T + \dfrac{r_2h_2^3}{\mu_2\Delta r^2}P_B + \dfrac{h_1^3}{r_i\mu_1\Delta\theta^2}P_R + \dfrac{h_3^3}{r_i\mu_3\Delta\theta^2}P_L + 12\pi r_i\left(\dfrac{R}{L}\right)^2\dfrac{h_1 - h_3}{\Delta\theta}}{\dfrac{r_2h_2^3}{\mu_2\Delta r^2} + \dfrac{r_4h_4^3}{\mu_4\Delta r^2} + \dfrac{h_1^3}{r_i\mu_1\Delta\theta^2} + \dfrac{h_3^3}{r_i\mu_3\Delta\theta^2}} \qquad [3]$$

In the case of thrust bearing, the pressure along all the boundary lines is equal to zero. In order to preserve this condition, the image points outside the boundary are equal in magnitude and opposite in sign to the value of pressure points inside the boundary.

Referring to Figure 1 in a 7 x 7 field:

P at 1 $= -P_B = -P_L$

P at 2–6 $= -P_L$

P at 7 $= -P_T = -P_L$

P at 8, 15, 22, 29, 36 $= -P_B$

P at 14, 21, 28, 35, 42 $= -P_T$

P at 43 $= -P_B = -P_R$

P at 44–48 $= -P_R$

P at 49 $= -P_T = -P_R$

Thus, the pressure at every point in the mesh is expressed in terms of the surrounding pressures, viscosities and film thicknesses. Employing the process of iteration, the pressure at every point in the mesh is determined. The Liebmann method of iteration was used in the pressure computation. The process of iteration is continued until the pressure converges to a prescribed error. The maximum error was specified to be less than 0.001.

$$\text{error} = \frac{\sum_m^m \sum_n^n |P'_k - P'_{k-1}|}{\sum_m^m \sum_n^n P'_k} \qquad [4]$$

where k = number of iterations.

Solutions were obtained for two film functions.

a) Michell (uniform film thickness in the radial direction)

Per unit film thickness is expressed as:

$$\frac{h}{b} = \frac{h_1}{b} + (1 - \theta_i) \qquad [5]$$

where h_1 = minimum film thickness

b = taper

b) Tapered land bearing (Refer to Figure 2)

Per unit film thickness is expressed as:

$$\frac{h}{h_1} = \frac{0.003\theta_i\left[3 - \dfrac{\Delta r_i}{L}\right]}{h_1} + 1 \qquad [6]$$

Force Computations

The force at a point is equal to the pressure at that point times the area. The total force is the integral of pressure times area summed over the whole bearing pad.

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Thus: $F = \sum P\,dA$

Since the form of a thrust bearing pad is a sector and the area of a sector is r $d\theta$ dr, the equation becomes:

$$F_v = \sum \sum P'_k r_i \, d\theta \, dr$$

$$F_v = \sum \sum P'_k r_i \, \Delta\theta \, \Delta r$$

$$F_v = \Delta\theta \, \Delta r \sum \sum P'_k r_i \qquad [7]$$

Analysis indicated that equal accuracy and fewer number of iterations were necessary in the calculation of the resultant force with a 7 x 7 field using trapezoidal integration than with a 12 x 12 field using step integration (Ref. 1). This observation was very important from the standpoint of efficiency for it resulted in approximately 60% cost and time reduction per calculation.

Several methods of integration, such as Simpson rule or Trapezoidal rule, may be used for total force calculation.

Double integration using the trapezoidal rule was adopted (Ref. 2). Using the trapezoidal method of integration, it is seen that all internal pressure points have the number 1 as a coefficient, the boundary points, except the corner points, have the number 3/4 as a coeffcient and the corner points have the number 9/16 as a coefficient. Thus, when the pressure field is modified by the operator indicated in Fig. 3, the value of force may be calculated by Equation 7. (Ref. Table I, Fig. 5).

Flow Computations

The hydrodynamic flow out of each edge of the sector was computed.

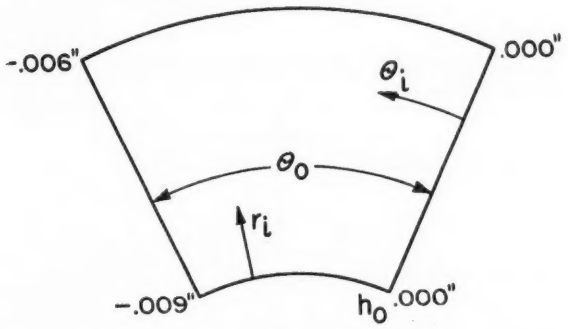

Fig. 2. Taper Dimensions of Tapered Land Thrust Bearing

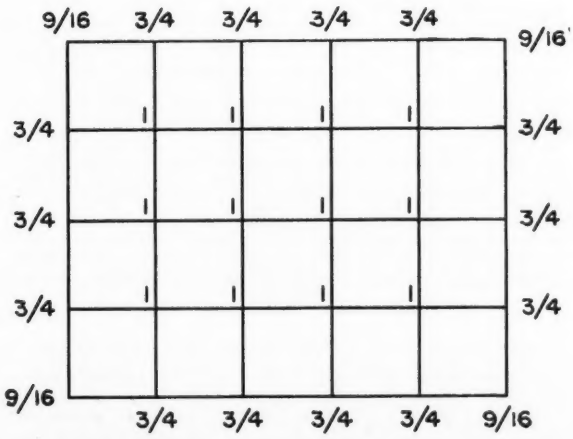

Fig. 3. Pressure Coefficients for Trapezoidal Integration

$$dQ_r = \frac{h^3}{12\mu} r \frac{\partial P}{\partial r} d\theta$$

$$dQ_\theta = \frac{h^3}{12\mu} \frac{1}{r} \frac{\partial P}{\partial \theta} dr$$

Now, defining a factor "q" as follows:

$$q = \frac{Q}{\frac{60}{231} \pi NRLb} \qquad \text{or } q = \frac{Q}{\frac{60}{231} \pi NRLh_1}$$

(depending upon film shape)
and rewriting we get

$$q_r = 0.0265 \frac{L}{R} \int_{\theta_i}^{\theta_o} \frac{h^3}{\mu} r \frac{\partial P}{\partial r} d\theta$$

$$q_\theta = 0.0265 \frac{L}{R} \int_{r_i}^{R} \frac{h^3}{\mu r} \frac{\partial P}{\partial \theta} dr \qquad [8]$$

where
Refer to Figure 4.

TABLE I. MICHELL BEARING

Θ_o = 0.69 radians

L/R = 0.5618

h_1/b	Error	F_v	k	Load Capacity — lb: IBM puter Com- Solution Solution	Michell Exact Solution	Approxi- mate Solution
1/6	0.000973	3.225504	21	293,000	241,000	397,000
2/6	0.001445	1.24706	20	113,500	112,400	125,000
5/6	0.000825	0.233964	15	21,000	20,750	21,000
1	0.000993	0.153035	24	13,900	———	12,500

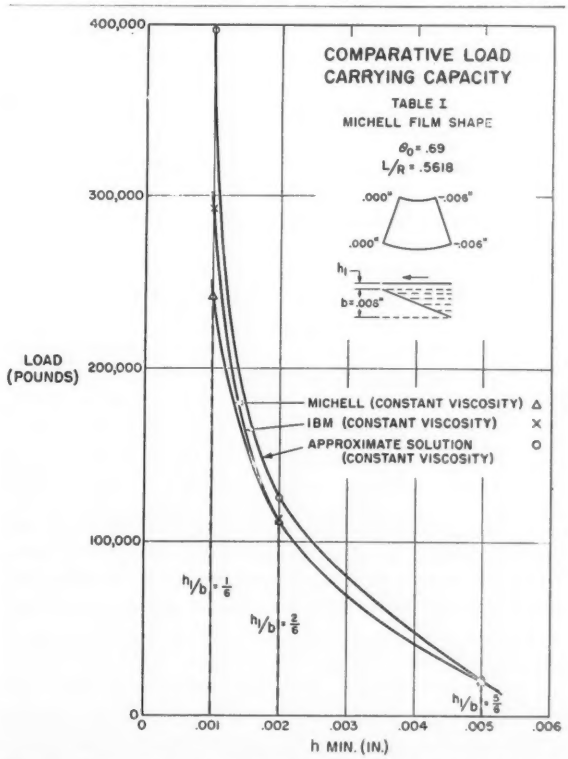

Fig. 5. Comparative Load Carrying Capacity for a "Michell" Bearing.

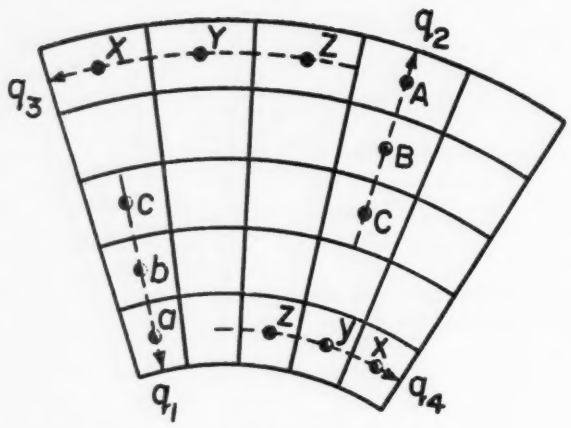

Fig. 4. Pressure Points Used to Determine Flow Rates

$$\left.\frac{\partial P}{\partial r}\right|_{\text{inner radius}} = \frac{1}{\Delta r}\,(3.75P_a - 0.8333P_b + 0.15P_c)$$

$$\left.\frac{\partial P}{\partial r}\right|_{\text{outer radius}} = \frac{1}{\Delta r}\,(3.75P_A - 0.8333P_B + 0.15P_C)$$

$$\left.\frac{\partial P}{\partial \theta}\right|_{\theta_{in}} = \frac{1}{\Delta \theta}\,(3.75P_X - 0.8333P_Y + 0.15P_Z)$$

$$\left.\frac{\partial P}{\partial \theta}\right|_{\theta_{out}} = \frac{1}{\Delta \theta}\,(3.75P_x - 0.8333P_y + 0.15P_z)$$

These slopes (Ref. 3) when substituted into Equations [8] yield Equations [9] which are listed below:

$$\text{flow at inner radius} = q_1 = 0.0265\,\frac{L}{R}\,\frac{\Delta\theta}{\Delta r}\sum^{m}\sum^{n}\frac{h^3 r}{\mu}\,(3.75P_a - 0.8333P_b + 0.15P_c)$$

$$\text{flow at outer radius} = q_2 = 0.0265\,\frac{L}{R}\,\frac{\Delta\theta}{\Delta r}\sum^{m}\sum^{n}\frac{h^3 r}{\mu}\,(3.75P_A - 0.8333P_B + 0.15P_C)$$

$$\text{flow at leading edge} = q_3 = 0.0265\,\frac{L}{R}\,\frac{\Delta r}{\Delta\theta}\sum^{m}\sum^{n}\frac{h^3}{\mu_r}\,(3.75P_X - 0.8333P_Y + 0.15P_Z)$$

$$\text{flow at trailing edge} = q_4 = 0.0265\,\frac{L}{R}\,\frac{\Delta r}{\Delta\theta}\sum^{m}\sum^{n}\frac{h^3}{\mu_r}\,(3.75P_x - 0.8333P_y + 0.15P_z) \qquad [9]$$

Results for the per unit flow are given in Table II.

TABLE II

h_1/b	q_1	q_2	q_3	q_4	Pivot Location % of Θ_o	Pivot Location % of L
1/6	0.1111	0.2719	0.3121	0.0805	73.70	56.25
2/6	0.1059	0.2670	0.2900	0.1213	67.40	55.20
5/6	0.1038	0.2635	0.2625	0.1620	61.05	54.30
1	0.1038	0.2634	0.2565	0.1680	58.80	54.60

Tests were conducted to determine temperature and pressure profiles as a function of minimum film thickness. Using the experimentally obtained temperature distribution, it was possible to convert it to a viscosity distribution. Then, employing the technique described above with the help of the IBM computer, the pressure distribution for the variable viscosity was calculated. The divergence between the two results is shown in Table III and Fig. 6. These graphs point out the need for the simultaneous solutions of Reynolds and energy equations; for, even if viscosity corresponding to outlet temperature is used in calculations, large errors in total load as a function of minimum film thickness result.

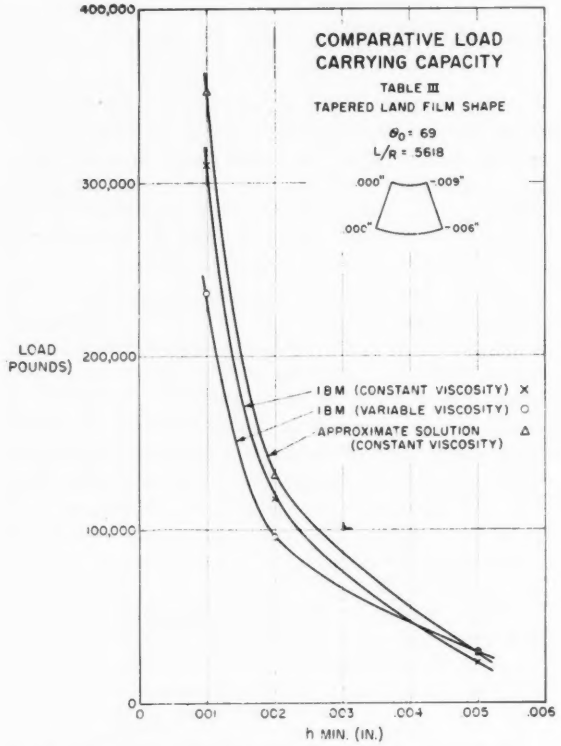

Fig. 6. Comparative Load Carrying Capacity for a "Tapered Land" Bearing.

TABLE III. TAPERED LAND BEARING

Θ_o = 0.69 radians
L/R = 0.5618

h_1	Error	F_v	k	Load Capacity — lb: IBM Computer Solution	k	IBM Computer Solution, Variable Viscosity	Approximate Solution
0.001	0.000848	0.095199	13	311,000	22	236,000	353,000
0.002	0.000752	0.144463	17	118,000	23	96,300	132,000
0.005	0.000713	0.174789	15	22,900	30	27,800	28,000

ANALYTICAL SOLUTIONS OF THE REYNOLDS' EQUATIONS FOR SECTOR THRUST BEARINGS

The tapered land thrust bearing has been solved for several film thickness configurations. Boswell (Ref. 4) has solved the Reynolds' equation for a sector having a taper

which depends only on the angular coordinate. The pad in this case has a warped surface. Solutions of the Reynolds' equations can be obtained for film thickness variations which are exponential in radial and angular coordinate directions (Ref. 5). In cases where the film thickness inlet-outlet ratios are not greater than 3 or 4, these exponential surfaces are reasonably close to planes and may be used as approximations for planes. In either case, the solutions are never obtained in a closed form, but rather in a rapidly convergent series of the Fourier type which can be carried out to any accuracy desired.

The Michell solutions were used here whenever applicable, but in most cases approximate solutions were used. (Ref. 5).

NUMERICAL COMPARISON

A numerical comparison was made of a tapered land bearing (tapered in radial and circumferential direction) and a Michell bearing (circumferential taper only). The comparison was made on a 273 sq in 8-pad bearing having an outer radius of 11.12″ and an inner radius of 4.87″. Each pad had a subtended arc of 0.69 radians. The operating speed of the bearing was 3600 rpm and the outlet oil viscosity was 1.41 x 10^{-6} lb-sec/in.^{2}

Table I and Fig. 5 compare three different methods employed in the calculations of the load-carrying capacity of Michell bearing. Table II gives solutions of "q" for the dimensionless pressure-induced flow across the edges of the pad. Included also are values of $\%\theta$ and %L (pivot location) which satisfy equilibrium of moments for the Michell bearing.

Table III and Fig. 6 compare the load-carrying capacity for tapered land bearing. Two methods of calculation are employed: 1. Using difference equations and employing the IBM computer; 2. Using exponential expression for film thickness (the approximate solution). These calculations are based on outlet oil viscosity of 1.41 x 10^{-6} lb-sec/in.^{2} The IBM computer was also used when variable oil viscosity was employed in the calculation of load-carrying capacity.

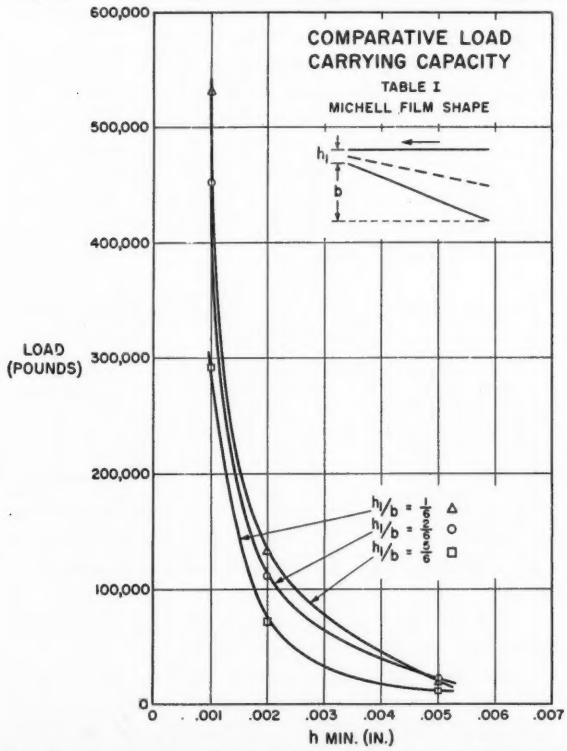

Fig. 7. Comparative Load Carrying Capacity of "Michell" Bearing as a function of pad inclination.

Fig. 7 compares the load-carrying capacity of Michell bearing operating with constant h_1/b ratio, which implies variable inclination of pad. Comparison can be made of the load-carrying capacity for the tapered land bearing and Michell bearing (Fig. 5 and 6).

REYNOLDS' MOMENTUM EQUATION IN CYLINDRICAL COORDINATES

$$r\frac{\partial}{\partial r}\left[\frac{h^3}{\mu}r\frac{\partial P}{\partial r}\right]+\frac{\partial}{\partial \theta}\left[\frac{h^3}{\mu}\frac{\partial P}{\partial \theta}\right]=6\omega r^2\frac{\partial h}{\partial \theta}$$

MICHELL'S SOLUTION

The equation is solved for the case where

$$h = h_1(1 + B\theta)$$

APPROXIMATE SOLUTION

The equation is solved for the case where

$$h = h_1 e^{(ar+b\theta)}$$

CONCLUSIONS

The following conclusions result from cases computed:

1. All three methods of computation yield comparable answers if the viscosity is considered constant and the minimum film thickness is not too small ($h_1 \geqq 0.0015$ inches).
2. The employment of exponential surface for the film shape yields reasonable results only when the inlet-outlet ratio is less than three. In any case, this method of solution is extremely laborious.
3. If the experimentally determined variable viscosity is taken into account, the load-carrying capacity prediction is considerably lower than that predicted by using a constant average viscosity. This is in agreement with experimental and field experience.
4. Digital computers seem to lend themselves to the solution of more realistically defined bearing problems and suggest ways in which more reliable performance predictions may be achieved.
5. Trapezoidal integration should be employed in order to reduce the cost and time per calculation.
6. The effect of variables is determined in a matter of minutes once the computer is programmed.

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3. Chung-Hua Wu, National Advisory Committee for Aeronautics; Technical Note 2214.
4. R. O. Boswall, "The Theory of Film Lubrication," pages 159-174.
5. Unpublished solution developed by the authors which is similar to the solutions by A. Charnes, E. Saibel and S. C. Ying, published in ASME Paper 52-A-34.

Acknowledgement

The authors wish to acknowledge the assistance of Miss Marie Conners and Mrs. Caroline Krueger whose patient care in performing the calculations made this paper possible.

A Rear Axle Problem: Initial Axle Noise

Initial axle noise — one of the oldest problems in the industry — is distinguished from noises associated with wear, since wear noises are of lower frequency. Initial noise is the objectionable sound heard in the passenger compartment at certain speeds and attributed to the axle.

All phases of the automotive and axle industries have at one time or another been involved in discussions on the causes of the noise. If the phenomenon is placed in its true position, a better understanding of the other axle problems should result. The material is not directly related to fuels and lubricants but should be of assistance to the oil industry in placing a proper perspective on initial noise in order to cope with the axle problems of wear and load carrying ability.

J. E. Cardillo
Ford Motor Co.
Dearborn, Mich.

The purpose of this report is to describe a laboratory method used in isolating the axle housing as one source of initial noise. The problem continues to plague the automotive divisions, the axle manufacturers, and the ultimate vehicle user.

Each of the automotive divisions has its own unique methods of desensitizing a vehicle, such as live body mounts, special body insulation, and rubber axle mounts. Each of the axle manufacturers has its own unique techniques to ensure quiet axles, such as modified cutting summaries, lap cycle, and pattern position. These pet methods and techniques, to mention only a few, are indicative of the confusion that exists.

The axle manufacturer is obligated to build an initially quiet axle. It is possible that wear life and load carrying ability are being sacrificed inadvertently in order to meet this obligation. It is possible that these problems would be simplified if the initial noise phenomenon were better understood.

The report will describe the laboratory method used in isolating the axle housing as a source of noise. Solutions currently being considered will also be outlined. If the interests of others can be stimulated, the increased effort should result in a complete understanding of the phenomenon and along with this, appreciable savings to the industry.

THE PROBLEM

A definition of initial axle noise may be redundant but may also serve to clarify the discussion of the problem.

Initial noise must be distinguished from noise associated with wear since wear noises are generally of lower frequency. Also, initial noise is inherent in new axles having no measurable dimensional deviations and defies control by print specifications.

Initial noise is the objectionable sound heard in the passenger compartment at certain speeds and attributed to the axle. It can be defined further as being of a constant frequency that is related to the frequency of the gear tooth impacts or their multiples.

The noise can be recorded on tape during a road test and analyzed for frequency and intensity in the laboratory. This has been accomplished by others and the techniques employed are common knowledge.

In the installation to be discussed, an initial noise frequency of 470 cps was measured at speeds of 38 and 76 mph. Figure 1 shows graphs made of the phenomenon by analyzing the tape recordings with a sound analyzer. Frequency definition was obtained by more accurate methods than indicated by the graphs. The analysis established that the frequency of the initial noise was 470 cps at both the 38 and 76 mph ranges.

The graphs of the phenomenon are very revealing and should be analyzed further. The peaks indicate that axle noise is tone-like in character and strongly suggest resonance of one or more components of the vehicle.

It appears that the majority of vehicle noises occur below 300 cps. Since axle noise occurs at higher frequencies and is almost a pure tone, it is not easily masked by the ambient vehicle noises. This, in part, explains why axle noise is easy to distinguish from background noise which approaches an 80-90 db ambient level. Preliminary data indicate that axle noise 20-30 db below the ambient level is very objectionable.

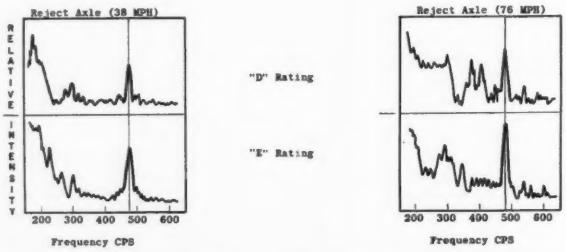

Fig. 1. Binaural Tape Recordings Road Test Procedure

Conclusions: The frequency of the noise heard is 470 cps at both the 38 and 76 mph ranges.

As stated previously, the tone-like qualities suggest resonance. In order to produce resonance, two conditions must be satisfied; namely, a vibrating body and an exciting force of the same frequency.

Figure 2 presents simple calculations which readily

Presented before the 12th Annual Meeting of the ASLE April 17, 1957 at Detroit, Mich.

established that at 76 mph, the frequency of the tooth impacts occurred at 470 cps, and at 38 mph, the frequency of the impacts occurred at 235 cps. It was concluded that the tooth impacts provided the exciting force for a noise producing component of the vehicle having a resonant frequency of 470 cps.

It is interesting to note that the 235 cps frequency is the half order sub-multiple of the fundamental frequency of 470 cps. This would indicate that the member in resonance is relatively undamped since excitation below the fundamental is possible.

The other important concepts to be concluded from these calculations are as follows:

1. An exciting frequency is present at all road speeds and will vary from 0 to 620 cps at 00-100 mph.
2. Impact forces will result during meshing even if the gear teeth are perfect.
3. These exciting forces may induce resonance of any car components whose natural frequencies are integral multiples of the exciting frequencies. At resonance, some components will produce noise and others will not depending on the frequency, the amplitude, and the ability of the component to move air.

Problem: Determine Frequency (F_T) of Tooth Impacts (cps) at Road Speeds of 38 & 76 mph.

Calculations:

$$F_T = \frac{N_S T_P}{60}$$

$$V = \frac{2\pi R \times T_G \times 60 \times N_S}{12 \times 5280 \times T_G}$$

$$N_S = \frac{528\ T_G V}{\pi R\ T_P}$$

$$F_T = \frac{528\ T_G V}{\pi R \times 60} = 6.2V$$

$$38F_T = 6.2 \times 38 = 235 \text{ cps}$$

$$76F_T = 6.2 \times 76 = 470 \text{ cps}$$

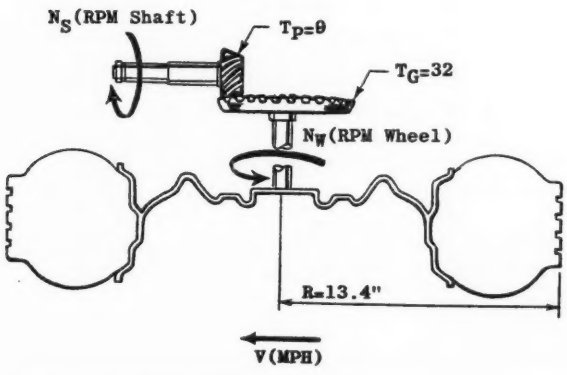

Fig. 2. Gear Tooth Impact Frequencies

Conclusions:

1. Tooth Impact Frequency at 76 mph is 470 cps..
2. Tooth Impact Frequency at 38 mph is 235 cps.
3. Tooth Impacts Appear to be the Exciting Force for the 470 cps Initial Noise Frequency.

It was reasoned further that if the conclusions were correct, it should be possible to induce resonance of this unknown noise-producing component in the laboratory by substituting an electro-mechanical vibrator for the gear tooth impacts.

It was anticipated that the noise-producing component would be found quickly under the laboratory conditions. However, during the initial surveys, the application of exciting forces at 235 and 470 cps induced vibration and noise of many components on the vehicle.

For this reason, it was decided to resort to trial and error road tests to localize areas under suspicion. The first step required removal of the body shell from the vehicle, and road tests established the presence of axle noise without the body. All further surveys were confined to the remaining parts.

In this interim period, the backing plates, the brakes, and the differential gears were removed successfully. Road tests again established the existence of axle noise. Having isolated the noise to the axle assembly, all additional work was confined to the laboratory.

The process of elimination was continued until all that remained was the rear axle housing. Figure 3 shows the laboratory set up that was used. Illustrated are the suspension, the vibrator, the microphone, and the sound analyzer.

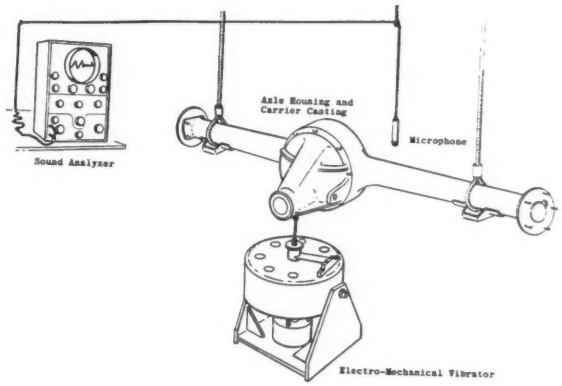

Fig. 3. Laboratory Set-Up

By applying the shaker to the nose of the carrier casting and exciting the complete housing at 235 and 470 cps, a resonance of 470 cps was detected by both the microphone and accelerometer pickups.

Figure 4 shows graphs of the microphone and accelerometer outputs applied to the sound analyzer.

The graph showing the shaker exciting the system at 235 cps is revealing in that the shaker microphone and accelerometer peaks are visible at 235 cps while the housing peaks occur at 470 cps.

These conditions coincide exactly with the road test results and confirm that the rear axle housing is primarily responsible for initial noise. If so, it can be concluded that the noise is primarily airborne or restricted to those paths in contact with the axle. It appears from preliminary results that very little noise is being transmitted through the spring seat pads.

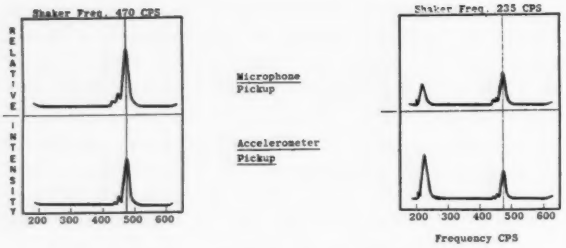

Fig. 4. Microphone and Accelerometer Pickups Laboratory Test Procedure

In a laboratory an Electro-Mechanical Vibrator was substituted for the gear tooth impacts. Both microphone and accelerometer pickups indicate that the housing will vibrate at resonance and produce noise at 470 cps when excited at either 235 or 470 cps.

The conclusions discussed can be better understood by a review of Figure 5. The sketch portrays a familiar type of pleasant noise producing instrument that closely parallels the initial axle noise phenomenon. The only unrealistic concepts in the illustration are as follows:

1. The shape of the axle housing — the violin case
2. The existence of a violin string
3. The existence of the plucking device on the ring gear which excites the string during rotation.

In reality, the artist's concepts are not far wrong. Remove the absurdities and all the elements of a violin exist. The gear tooth impacts in a conventional installation provide the same excitation directly to the housing as the string and the plucking device do in the illustration.

POSSIBLE SOLUTIONS

The problem does not appear simple. Some thought and discussion will indicate logical areas to be pursued.

A direct solution to a resonance problem would be to detune the system by making the natural frequency below or beyond the operating frequencies. In this case, the operating frequencies would necessitate a re-design of the housing to raise the fundamental from 470 cps to an estimated 1500 cps or to lower it to an estimated 150 cps. Either approach would present serious structural problems.

Another solution would be to reduce the force of the exciting frequency. This is in fact, what the axle manufacturers try to do by lapping, rolling and pattern positioning. Further refinements in this area are costly and impractical since impact forces will result during meshing even if the gears are perfect. It appears more logical to make the housing insensitive to resonance and then, possibly, greater gear tolerances could be allowed.

The only practical solutions appear to be either a damper or an absorber. The cost implications of the damper make it appear less promising at this time. The absorber principle appears interesting and work is progressing along these lines. In order to simplify the design, it will be necessary to define the vibration mode, possibly simplify it by modifications to the housing, and then devise an absorber that will have the same frequency as the housing but incapable of producing noise. The absorber is illustrated in theory in Figure 6.

Ultimately it may be possible to re-design existing components such as the springs, backing plates, etc., to act as an absorber and thus, obviating a separate device.

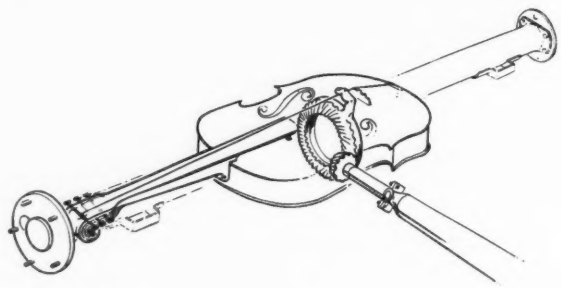

Fig. 5. Artist's Concept

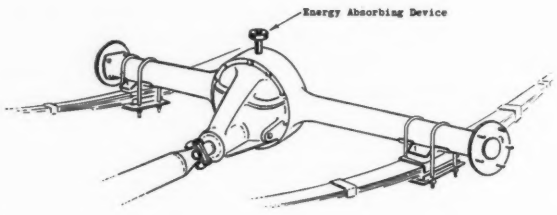

Fig. 6. Theoretical Energy Absorbing Device

CONCLUSIONS

Initial noise has been isolated to the axle housing and further development should lead to a practical solution. If its occurrence can be prevented or minimized, savings can be realized in the following areas:

1. *Gear manufacturing:* Increase tolerances, decrease lapping, minimize pattern position, and decrease rejects for sound characteristics.
2. *Axle:* Increase life and load carrying ability.
3. *Vehicle:* Eliminate rubber spring seat pads, eliminate driveshaft liners, minimize the importance of body mounts and body insulation.
4. *Owner complaints:* Minimize warranty and policy costs.

The areas specified are definite objectives that may be realized by pursuing the problem to a successful conclusion.

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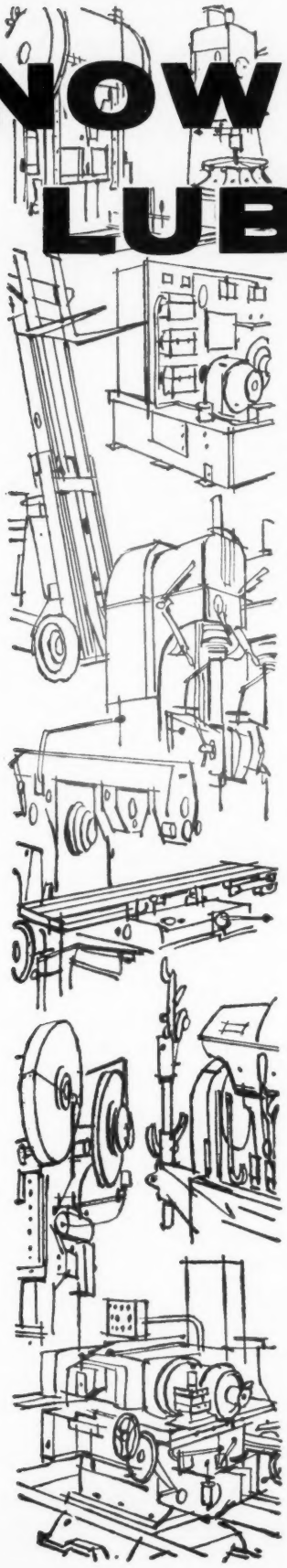

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ASLE News Notebook

EXECUTIVE REPORTS

At the June 4, 1957 Board of Directors meeting, the proposal to organize a new Southern Region was discussed. It was indicated that such a region should not presently be organized, pending the development of more interest in the South.

At a time when such a region becomes advisable, the report continued, it is suggested that two regions be organized, dividing the South into Southwest and Southeast. This would reduce some of the responsibilities of the Midwest Section, which now includes Sections as far away as Texas. It was recommended that this proposal be tabled for the present.

In recognition of the need for more representation in this area, however, W. E. Hamilton of The U. S. Steel Co., Fairfield, Ala. is being approached as an area representative in the Southern Area of the Central Region.

Midwest Region

All sections in the Midwest Region have been requested to submit a list of Sectional officers for the coming year, and to appoint a membership chairman for new members in the Section area. Each Section has also been advised to submit all material pertaining to his office to the incoming officer. It is expected that forecasts will be obtained from each section in the near future regarding new members.

All five area representatives for the Midwest Region have agreed to serve in the same capacity for another year.

On June 28, an organizational meeting was held in the Tri-Cities Area. E. W. Brennan, Pure Oil Co., Crystal Lake, Ill. was the speaker.

A. B. Wilder has been appointed chairman of a committee composed of all regional vice-presidents to compile a complete outline of duties of various Section officers, and to forward this information to all local Sections.

M. E. Dougherty was appointed chairman of a similar committee to establish a suitable report form for Area Representatives to advise regional vice presidents of their activities.

One hundred and fifty students enrolled in ASLE Connecticut Section's practical lubrication course view problems first-hand at Scoville Mfg. Co.'s continuous strip mill.

COMMITTEE REPORTS

A committee consisting of the following members was appointed to consider revision of the by-law structure: L. B. Sargent, Aluminum Company of America; R. L. Humphreys, Bethlehem Steel Co.; P. F. Chappell, American Oil Co.; M. E. Merchant, Cincinnati Milling Machine Co.; A. E. Cichelli, Bethlehem Steel Co.; C. L. Willey, ASLE Administrative Secretary. Messrs. Merchant and Willey are serving in advisory capacities.

On May 29, a meeting was held in Bethlehem, Pa. to begin rewriting the by-laws along the lines approved at the board meeting in April. After the revision has been completed by the group, it will be presented to the board for approval and submission to the vote of the membership. The Committee hopes to complete its assignment within a year.

Lubrication Conference Committee

Programming has been confirmed for the fourth annual ASLE-ASME joint Lubrication Conference, to be held Oct. 7, 8, 9 at the Royal York Hotel, Toronto, Ontario, Canada. Session topics scheduled include: Recent Experiments In Wear, *Chairman*, A. C. West, California Research Corp.; Boundary Lubrication, *Chairman*, E. Rabinowicz, MIT; High-Temperature Lubrication, *Chairman*, A. Hundere, Southwest Research Institute; Factors Affecting Contact Fatigue, *Chairman*, E. G. Jackson, General Electric Co.; Fluid Film Lubricated Bearings, *Chairman*, E. A. Allen, University of Toronto; Recent Studies of Fluid Film Lubrication. *Chairman*, F. R. Archibald, Arthur D. Little Co.

Two Reporter Sessions will be included. In all, 20 research papers will be presented at the various sessions.

The following appointments to the 1958 Lubrication Conference Committee have been confirmed: A. C. West (Conference Chairman and Technical Papers Coordinator), A. Hundere, E. G. Jackson, and R. L. Johnson.

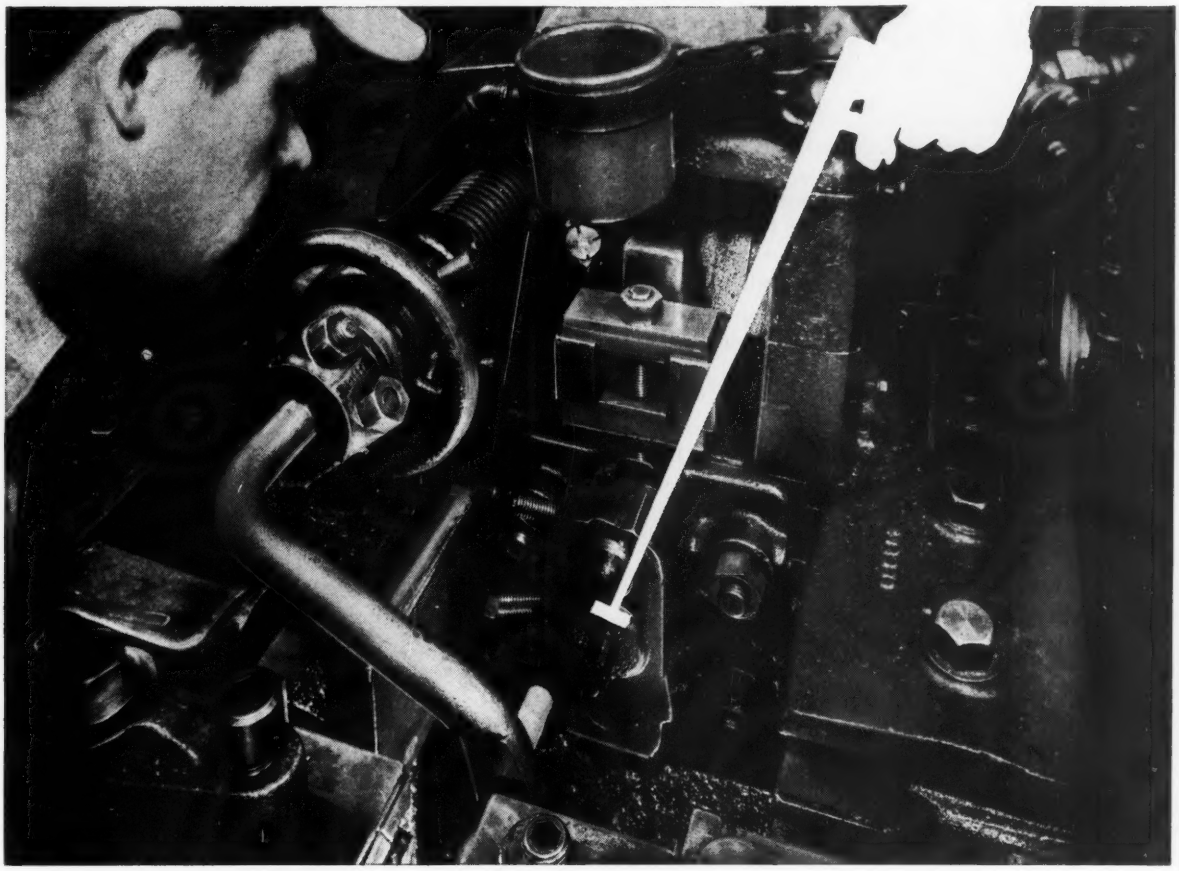
Membership and Admissions

Total membership count as of April 30 — 2828 including all classifications of membership. Broken down, the count includes 2684 regular, 71 sectional sustaining, 28 associated, seven honorary, four students, and 34 industrial.

Three-hundred and seven members were dropped from the roster during April for non-payment of dues, resig-

(continued on p. 470)

A Touch here does so much



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SECTION NEWS....

Boston Section—Final section meeting of season (May 20), past chairman's night. Revote on nominated slate of 1957-8 officers resulted in reconfirmation—Secretary requested to cast one unanimous vote as before. Thirty-five members and guests viewed "Fire Power," an Ethyl Corporation scientific display of gasoline power . . .

Kingsport Section—Steak dinner meeting (May 24) highlighted by a talk by Oak Ridge's Tell Martin on 12th annual ASLE meeting. After talk, a 45-minute tour of Belding Hemingway's new plant in Morristown, Tenn. . . .

Los Angeles Section—June 12 meeting featured annual ladies' night and behind-the-scenes-visit to L.A.'s Hotel Statler. Members and wives peered at banquet kitchens, commissary, laundry, carpet cleaners, and engine room and inspected typical rooms and special suites of this "city within a city." . . .

Cleveland Section—"Stump the Experts Nite" played to 42 members and guests. Four "experts" (R. H. Josephson, Cleveland Graphite Bronze Co.; J. Hurley, Ford Motor Co.; Dr. W. E. Campbell, Consultant; Mike Papp, Warner & Swasey Co.) were quizzed on audience-submitted questions. Judges Frank M. Aldridge (Aldridge Industrial Oils), Dr. Peter A. Asseff (Lubrizol Corp.), and Prof. R. R. Slaymaker (Case Institute of Technology) evaluated answers, gave silver dollars to questioners who stumped experts . . .

Pittsburgh Section—At the May 9 meeting, Dr. G. K. Bhat, research metallurgist at Crucible Steel Co. of America spoke on "A Metallurgist Looks at Lubrication Problems in Metal Working and Aircraft Bearings." . . .

Connecticut Section—First course on practical lubrication a success—six session course attended by about 75 people, many non-members from local industries. Basic principles, tests and manufacture of oils and greases, antifriction and plain bearings, hydraulic systems, gears and methods of application were discussed by instructors from Esso, Socony, Gulf, and Shell Oil Companies, New Departure Div. of General Motors, Farrel-Birmingham, and Lincoln Engineering Co. Final session of the course was held in conjunction with a "Roll Oil Panel" meeting, including a 150-man tour of Scovill Mfg. Co.'s continuous strip mill . . .

Elections—New officers were elected by the Philadelphia, St. Louis, Milwaukee, Montreal, Ontario, Twin Cities, and New York Sections. (See Directory) . . .

NEWS NOTEBOOK

(continued from p. 468)

nations, deaths, etc. Some of those dropped are expected to renew by making late payment of dues. It is believed that a still greater number could be induced to renew if better local follow-up were practiced.

An aggressive membership incentive plan for 1957-58 was outlined by B. M. Dunham, membership chairman. It involves establishment of a local Section "Target List", whereby local Sections would forward names of companies in their areas to National Headquarters for membership invitation. Other points under this plan include establishing representatives in large industrial plants, increasing personal contacts, forwarding Section announcements to non-members, encouraging in-Section educational courses, membership goals.

Industrial Committees

Under the direction of E. J. Gesdorf, planning for the 1958 Cleveland Convention is well under way. On May 2, officers of the Cleveland Section, and chairmen for the various committees were selected: Sessions Coordination, T. J. Walsh; Finance, C. C. Herrod; Publicity, E. W. Baumgardner; Entertainment, P. A. Asseff; Welcoming, W. E. Campbell; Ladies, Mrs. W. E. Campbell.

On May 28, Mr. Gesdorf called another committee meeting attended by C. L. Willey, J. A. Brady, C. F. Cowley, and W. J. Wojtowicz, plus the newly appointed committee chairmen. Another meeting was held June 25 to establish a budget for 1958 convention expenses to be submitted to the National Finance Committee.

Canadian Region

On May 14, the Ontario Section completed a drive for new members. Thirty-five new applications were forwarded to the National Office. The Montreal Section is also presently developing plans for a new member drive next autumn. As an incentive proposal, T. A. Marshall, Canadian vice president, suggested a short lubrication engineering course at McGill University.

Election of the Nominations Committee at the June 4 meeting resulted in the unanimous appointment of J. W. Hopkinson and J. Boyd. E. R. Booser, past-president, is automatically chairman of this committee.

The name of the Technical Committee on Lubricant Reclamation and Disposal was changed to Technical Committee on Lubricant Reconditioning and Disposal.

Other Motions

J. H. Fuller, D. M. Cleaveland: That proposal made by B. T. Harding for rebating a portion of the dues from Industrial Members to local Sections be tabled until the October meeting. Motion carried.

D. W. Sawyer, J. D. Lykins: That the Journal Bearing Abstracts be published. Motion carried.

J. L. Finkelmann and J. D. Lykins: That, if the Gear Committee holds a symposium, the Society should publish the papers in a single bound volume. Motion carried.

J. D. Lykins, A. B. Two: That the Finance Committee be authorized to withdraw funds to finance Cleveland section expenses for the 1958 Annual Meeting based on a budget to be submitted at an early date by the Cleveland Section for Finance Committee approval. Motion carried.

SPECTROCHEMICAL ANALYSIS HANDBOOK — Hilger & Watts Ltd. offers a 24-page brochure describing design and applications of a variety of excitation source units and discharge stands used in spectrochemical analysis. This illustrated brochure includes detailed specifications of arc and spark units, stands, attachments, and accessories available for both direct reading and photographic analysis.

The publication is the second in a series of spectrochemical analysis handbooks and will be distributed in the United States by the Information Service of the Jarrell-Ash Co., 26 Farwell St., Newtonville 60, Mass.

ASLE Preprints Available

(from 12th Annual Meeting)

A limited supply of the following Preprints of papers presented at the ASLE 1957 Annual Meeting are available at 35c each to members, 50c each to non-members. To order, indicate the number of copies desired, fill in your name and address, enclose remittance, and mail to:

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____(57AM-5B-2) *Car Oil Additives,* by L. C. Atchison
____(57AM-5A-5) *Chemical Additives for Hypoid Gears,* by C. V. Smalheer
____(57AM-4A-1) *Chemical Nature of Extreme-Pressure Lubrication, The,* by G. Hugel
____(57AM-2A-1) *Choose Your Press Drawing Lubricants with Care,* by L. Salz
____(57AM-6A-4) *Circulating Oil Systems from 1/50 to 50 GPM,* by E. W. Baumgardner
____(57AM-6A-2) *Considerations in Filter Selection,* by H. A. Wilson
____(57AM-2A-2) *Conversion Coatings Aid the Lubrication of Titanium,* by E. L. White & P. D. Miller
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____(57AM-2A-4) *Threading Test for Cutting Oil Evaluation, A,* by H. A. Hartung, J. W. Johnson & A. C. Smith, Jr.
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____(57AM-6B-1) *Wear & Frictional Characteristics of Some Wrought & Cast Nickel-Base Alloys,* by R. A. Kozlik
____(57AM-1A-1) *Wear of Carbon-Type Seal Materials with Varied Graphite Content,* by M. A. Swikert & R. L. Johnson

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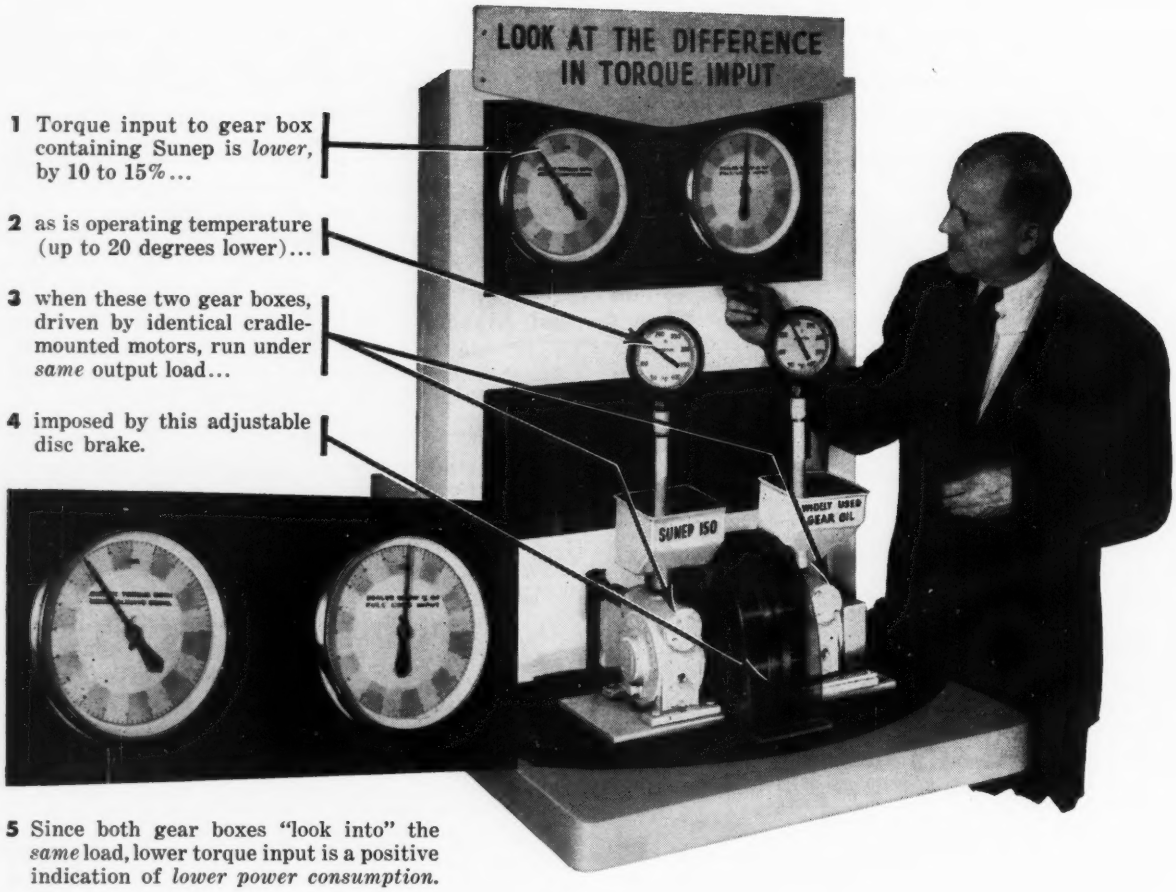
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PERSONALS...

Four new appointments further expanding the sales service facilities of the recently established Chemical Division of Catalin Corporation of America have been announced by Harry Krehbiel, president.

John Ziccarelli was named to Catalin's New York office; Herman Pickles, Jr., was assigned to the company's Chicago office, and E. B. Lee to the Houston, Texas office.

★ ★ ★

T. J. Coleman has been appointed vice-president of Union Carbide Development Company.

★ ★ ★

Fred G. Moser has been promoted to manager of original equipment sales by Johnson Bronze Co., producers of sleeve bearings and bushings, with offices at New Castle, Pa.

★ ★ ★

The appointment of the Becker Engineering & Supply Company of Chicago as a new Manzel representative has been announced by Robert W. Meyers, Manzel sales manager. In its new capacity, the Becker Company will handle all sales for Manzel's line of force feed lubricators, chemical feeders and slurry pumps in the industrial areas of Illinois and Indiana.

★ ★ ★

The Marrick Co. of Louisville, Ky. has been appointed exclusive sales representative for Tuthill Pump Co., Chicago, Ill., manufacturers of rotary gear pumps.

★ ★ ★

The Lubrication Engineering profession, the Metals Industry, and the Oil Industry, were grieved to learn of the passing of *Arthur Lieberman* on March 31, 1957, after a long illness.

Mr. Lieberman was born March 24, 1903, in Chicago, Illinois. He received his degree in Mechanical Engineering from Northwestern University.

Before establishing his own business, the Petroleum Marketing Company, in 1940, he was associated with the Quaker State Oil Company and Barnsdall Refining Company as a lubrication engineer.

He was an active member of many professional societies including A.S.L.E., A.S.M., A.I.C., A.C.S., A.S.M.E., and the Chemical National Research Association, and was a registered California Professional Engineer in Mechanical Engineering.

★ ★ ★

PHIL D. YATES

Two new appointments have been made by the Garlock Packing Co., Palmyra, N. Y. Phil D. Yates, formerly sales representative in Western New York and a Garlock 15-year-man, has been named district manager of the Pittsburgh branch of the firm. V. L. Moore is Garlock's new sales manager for the territory encompassing Los Angeles County and vicinity.

V. L. MOORE

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Membership in the American Society of Lubrication Engineers is in several grades as defined below. Assignment to grade is made by the Admissions Committee or Board of Directors on the basis of information submitted or supplied by references.

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Associate Members: Associate Members shall be persons less than 24 years of age, and those who do not completely fulfill the membership requirements for Members. Fee $7.50.

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Foreign Lubrication Abstracts

Reactions Involved in the Impregnation of Steel Surfaces with Metals. V. A. Illarionov. METALLOVEDENIE I OBRABOTKA METALLOV, March 1957, No. 3, pp. 2-8; 3 figures, 1 table, 3200 words. [Reactions underlying metallic surface impregnation of steel by formation of a volatile metal compound (for example, chloride) on steel surface and chemical reaction therewith, accompanied by enrichment of steel surface with the particular metal. Literature data on three possible types of reaction and reasons why they cannot be correct. Author's experiments on pack and gas aluminizing to study the mechanism of chemical transfer of aluminum. Results and their interpretation: role of autoreduction-autooxidation reactions connected with formation of lower-valency metal compounds. Unified explanation of (and theory for) aluminizing, siliconizing, chromizing, and carburizing.] (Order No. 6967, price $7.75)

Influence of Lubricants upon the Surface Flow of Metal. S. Ya. Veiler. DOKLADY AKADEMII NAUK SSSR, vol. 99, 1954 #p, pp. 1025-27; 2 figures, 1400 words. [Investigation of influence of different lubricants upon tangential force, microstructure, and microhardness through flow in surface zone of metals. Experimental setup and procedure. Lubricants used in study: 5% soap, soln., water; octane; octyl alcohol; oleic acid; also dry. Surface-active lubricants as plasticizers of metal surfaces; microstructure of copper (Fig. 2) after deformation with and without surface-active lubricant. Analogy between mechanism of action of lubricant in surface deformation and that in bulk deformation.] (Order No. 3519, price $2.80)

New Methods of Heat Treatment and Surface Impregnation of Metals in Fused Salts. A. I. Zot'ev. VESTNIK MASHINO-STROENIYA, vol 35, 1955, #5, pp. 67-71; 4200 words. [Highlights of papers read at a conference on heat treating of steel and nonferrous metals in fused salts and alkalies. Subjects covered include sulfurizing and its numerous applications; need for further development work on sulfurizing techniques, bath compositions, and the factors governing the wear resistance of sulfurized parts. Effect of adding 5-10% sodium sulfide to conventional cyaniding baths.] (Order No. 3955, price $8.40)

Mechanism of Action of Lubricants in the Working of Metals. S. Ya. Veiler, V. I. Likhtman, and P. A. Rebinder. DOKLADY AKADEMII NAUK SSSR, vol. 110, 1956 #6, pp. 985-988; 4 figures, 2100 words. [Interpretation of action of lubricants in metal working in terms of reduction of thickness and build-up ('plastic wave') in front of the tool, and of rendering surface layers more amenable to plastic deformation. Specific advantages of surface-active lubricants in regard to their action on surface of metal, its elastic recovery, stress state, and surface quality. Examples: forcing a metal ball thru an opening in sizing (Figs. 1 and 2); wire drawing (Fig. 3); and deep drawing (Fig. 4).] (Order No. 3943, price $4.90)

The Problem of Graphitization of Low-Carbon Steel. By I. V. Salli and I. I. Pyasetskii. FIZIKA METALLOV I METALLOVEDENIE, vol. 3, 1956, #3, pp. 513-515; 3 figures, 1 table, 1300 words. [Report on experiments intended to throw light on the mechanism of graphitization of low carbon steel, with special reference to processes occurring during heat treatment in vacuum. Composition of steels used in study (Table 1). Effect of cooling rate upon **precipitation of graphite on specimen surface.** Interpretation of results with the aid of the ferrite-graphite equilibrium curve (Fig. 3) in the iron-carbon phase diagram: causes of graphitization of low carbon steel.] (Order No. 3952, price $3.35)

(This is the twelfth in a series of abstracts of translations of foreign articles on lubrication, by Henry Brutcher. Copies of the full translations are available, at the indicated prices, by writing Mr. Brutcher at P. O. Box 157, Altadena, Calif.)

ASLE Preprints Available

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_____(56LC-7) *Application of Reaction Rate Theory to the Weld Junction Aspect of Kinetic Friction Between Unlubricated Metallic Surfaces,* by F. F. Ling & E. Saible

_____(56LC-1) *Application Problems with Petroleum Lubricants in Nuclear Power Plants,* by R. F. Hausman & E. R. Booser

_____(56LC-5) *Effects of Nuclear Radiation on Lubricants,* by J. A. King & W. L. R. Rice

_____(56LC-9) *Fretting of Hardened Steel in Oil,* by J. R. McDowell

_____(56LC-6) *Fundamental Study of Synthetic Sapphire as a Bearing Material, A,* by L. F. Coffin, Jr.

_____(56LC-2) *General Radiation Damage Problems for Lubricant & Bearing-Type Materials,* by V. P. Calkins & C. G. Collins

_____(56LC-18) *Influence of Temperature on Boundary Lubrication,* by C. W. Cowley, D. J. Ultee & C. W. West

_____(56LC-13) *Investigation of "Melt Lubrication,"* by B. Sternlicht & H. Apkarian

_____(56LC-10) *PbO & Other Metal Oxides as Solid Lubricants for Temperatures to 1000° F.,* by M. B. Peterson & R. L. Johnson

_____(56LC-4) *Radiation-Resistant Greases,* by J. G. Carroll, R. O. Bolt & B. W. Hotten

_____(56LC-8) *Solid Film Lubricant—Factors Influencing Their Mechanism of Friction & Wear,* by R. E. Crump

_____(56LC-12) *Solution for the Finite Journal Bearing & Its Application to Analysis & Design — II, A,* by A. A. Raimondi & J. Boyd

_____(56LC-16) *Specific Heat at Low Temperatures and Latent Heat of Fusion of Aircraft Engine Lubricants,* by V. E. Shrock & R. E. Gott

_____(56LC-11) *Temperature Effects in Journal Bearing Lubrication,* by W. F. Hughes & J. F. Osterle

_____(56LC-17) *Thermal Conductivity & Thermal Diffusivity of Aircraft Engine Lubricants at Low Temperatures,* by V. E. Schrock & E. S. Starkman

_____(56LC-3) *Use of Rolling Contact Bearings in Low Viscosity Liquid Metal Lubricants,* by W. Markert, Jr. & K. M. Ferguson

_____(56LC-14) *Yield Stress as a Factor in the Performance of Greases,* by D. Evans, J. F. Hutton & J. B. Matthews

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Summaries of Foreign Papers

(continued from p. 451)

of metallographic sections of the copper surface layers which were brightened in phosphoric acid and etched in chromic acid solution. Microhardnesses at various depths within the copper surface were measured also. This aspect of the investigation reveals that the degree of deformation and the microhardness are greatest at the surface and decrease as the measurement location penetrates the surface. In the case of the unlubricated metal, the grains in the surface layer show marked plastic flow and grain boundaries become indistinct. The grains appear as elongated thin plates. In the experiments with lubricants, the grains are less deformed and grain boundaries remain distinct.

TABLE I—PENETRATION FORCES

Lubricant	Maximum tangential force for penetration, in kg.
None (dry)	1900
Water	1800
Octane	980
Octyl Alcohol	750
Oleic Acid	680
Oxidized Paraffin Wax	650
Caprylic Acid	620
5% Soap Solution	540

TABLE II. ELASTIC RECOVERY

Lubricant	Difference in diam. of hole and ball, mm.	Thickness of squeezed layer, mm.	Elastic recovery, mm.
None (dry)	0.316	$h_o = 0.125$	$y_o = 0.191$
Oxidized Paraffine Wax	0.316	$h_1 = 0.025$	$y_1 = 0.291$

The author suggests that during pressure working of the metal an active lubricant plasticizes a very thin surface layer to form a soft metal film. This film can undergo heavy plastic deformation and acts as a solid lubricant. As a result, there is slipping between the steel ball and the copper metal. Reviewed by E. H. Loeser

"Effectiveness of The Action of Media in Cutting of Noble Metals," by N. A. Pleteneva and G. I. Epifanov, *Doklady Akademii Nauk, USSR, 110*, No. 3, pp. 414-416, 1956.

The effectiveness of the action of liquids in the cutting of metals is assumed to depend partly on the reduction of friction between the chip and the tool as the result of formation of surface films. Bowden, Gregory, and Tabor[1] have shown that film formation between the liquid and the metal markedly depends upon the nature of the metal. Inert metals would not be expected to respond to chemically active liquids. However, where lubrication is dependent upon physical adsorption, hydrocarbon chain length would be expected to have an effect.

The objective of the investigation was to determine the effectiveness of liquids in the cutting of noble metals such as silver, gold, and platinum, the least chemically active metals. Cutting experiments (free planing) were conducted with a tool set at a cutting angle of 58° and with a depth of cutting of 0.2 mm. Annealed samples of silver, gold, and platinum were used. Nonpolar hydrocarbons, water, and various representative homologous series of alcohols, acids, and esters of the aliphatic series, as well as carbon tetrachloride, were investigated.

The effectiveness of the liquid is expressed as the ratio of work of dry or unlubricated planing (A_o) to the work by planing in the given liquid (A). This ratio shows how many times the cutting work is reduced with the active liquid as compared to the work with unlubricated cutting. The data obtained are presented in the following table.

$$\text{Ratio } \left(\frac{A_o}{A}\right) = \frac{\text{Work of unlubricated planing}}{\text{Work of planing in given liquid}}$$

		Ag	Au	Pt
Nonpolar hydrocarbons		1.6	1.3	2.8
Water		1.4	1.2	2.4
Alcohols:	Methyl	1.6	1.4	2.3
	Ethyl	1.6	1.4	1.9
	Butyl	1.6	1.3	1.6
	Octyl	1.6	1.2	1.8
Acids:	Acetic	1.3	1.2	3.5
	Propionic	1.5	....	3.1
	Valeric	1.5	1.2	2.8
	Nonanoic or Pelargonic	1.5	1.2	3.0
	Oleic	1.4	1.1	2.0
Complex esters:	Ethyl propionate	1.6	1.4	2.9
	Ethyl pelargonate	1.5	1.3	3.1
	Ethyl laurate	1.6	1.3	3.2
Carbon tetrachloride		1.3	1.2	4.1

The table shows that the effect of all the various liquids is similar for silver and gold. The ratio is relatively low; i.e., not higher than 1.6. However, the liquids have a marked effect during the cutting of platinum. Nonpolar hydrocarbons, acids, and esters reduce the work of cutting platinum about three times, and carbon tetrachloride reduces it four times. Although not shown in the table above, nitrogen-containing compounds, such as nitrobenzene and aniline, reduce the work of cutting platinum six or seven times.

In the cutting of these metals, no characteristic influence of the length of hydrocarbon chain is observed. The ratio does not change with chain length in cutting silver in alcohols or silver and gold in acids and esters. However, the ratio decreases with increasing chain length when cutting platinum in alcohols, and drops very sharply with increased chain length when cutting platinum in acid. Thus, on the basis of the data, it appears that the effectiveness of the cutting liquid does not stand in a simple relationship with its chemical activity.

On the other hand the experimental work shows that the effectiveness of surface active liquids is a function of the concentration of the active component. In all cases a continuous increase in the efficiency of cutting is observed with increase in concentrations (in the order of tens of per cent). These concentrations do not correspond to saturation of the adsorbed layer.

The effectiveness of the liquid is higher in the case of platinum than it is for metals that are considerably more chemically active[2]. The low effectiveness of the liquids on silver and gold as compared with platinum is not explained by their chemical inertness, which is even lower than that of platinum, but by the mechanical properties of these metals.

The results are consistent with the authors' theory[3] that a metal is hardened during deformation because of

the diffusion of atoms into the lattice of the metal. The atoms are formed by catalytic decomposition of the adsorbed liquid on the freshly cut metallic surface. An indirect confirmation of the theory appears to be that, in the group of metals investigated, the effectiveness of the liquid correlates with the metal's catalytic activity, rather than its chemical activity.

REFERENCES

[1]Bowden, F. P., Gregory, N. N., Tabor, D., *Nature, 156* p 97 (1945); Bowden, F. P., Tabor, D., "Friction and Lubrication of Solids," Oxford University Press, 1950 (Russian translation in a handbook, "Friction and Boundary Lubrication," edited by I. V. Kragelsky, p 165, IL, 1953).

[2]Pleteneva, N. A., Epifanov, G. I., *Doklady Akademii Nauk, 77,* No. 6 (1951).

[3]Epifanov, G. I., Pleteneva, N. A., Rebinder, P. A., *Doklady Akademii Nauk, 97,* No. 2 (1954).

Reviewed by Douglas Godfrey

Coming Events

September 3-4—International Union of Geodesy and Geophysics (General Assembly), University of Toronto, Toronto, Canada.

3-6—Calorimetry Conference, Portsmouth, N. H.

5-7—American Physical Society (Regional Meeting), Boulder, Colo.

8-13—American Chemical Society (National Meeting), New York City.

9-12—Society of Automotive Engineers (Tractor Meeting & Production Forum), Hotel Schroeder, Milwaukee, Wisc.

9-13—American Society of Mechanical Engineers (Instruments & Industrial Regulators Div., Joint Conference & Exhibit with Instrument Society of America), Auditorium, Cleveland, Ohio.

9-13—Instrument Society of America (12th Annual Instrument-Automation Conference & Exhibit), Auditorium, Cleveland, Ohio.

9-13—New York University College of Engineering (Titanium Lecture Program), N. Y. U., New York City.

10-11—Petroleum Institute (Petroleum Packaging Committee), Point Edward, Ontario, Canada.

10-13—American Statistical Association (Annual Meeting of Section on Physical & Engineering Sciences), Atlantic City, N. J.

11-13—Canadian Association for Applied Spectroscopy (Symposium), Victoria Museum, Ottawa, Canada.

14-16—Armour Research Foundation (Industrial Nuclear Technology Conference), Museum of Science & Industry, Chicago, Ill.

15-18—American Institute of Chemical Engineers (Regional Meeting), Lord Baltimore Hotel, Baltimore, Md.

22-25—American Society of Mechanical Engineers (Petroleum Mechanical Engineering Conference), Mayo Hotel, Tulsa, Okla.

23-25—American Society of Mechanical Engineers (Fall Meeting), Statler Hotel, Hartford, Conn.

23-25—Standards Engineers Society (6th Annual Convention), Commodore Hotel, New York City.

23-26—Association of Iron & Steel Engineers (Annual Convention), Penn Sheraton Hotel, Pittsburgh, Pa.

24-25—Institute of Radio Engineers, and American Institute of Electrical Engineers (Industrial Electronics Conference), Morrison Hotel, Chicago, Ill.

30 thru October 2—American Oil Chemists' Society (Fall Meeting), Netherland Hilton Hotel, Cincinnati, Ohio.

October 1-4—National Association of Corrosion Engineers (North Central Region), Sherman Hotel, Chicago, Ill.

1-5—Society of Automotive Engineers (Aeronautic Meeting, Aircraft Production Forum & Aircraft Engineering Display), Ambassador Hotel, Los Angeles, Calif.

6-9—American Institute of Mining, Metallurgical & Petroleum Engineers (Petroleum Branch Meeting), Dallas, Tex.

6-10—Electrochemical Society (112th Meeting), Statler Hotel, Buffalo, N. Y.

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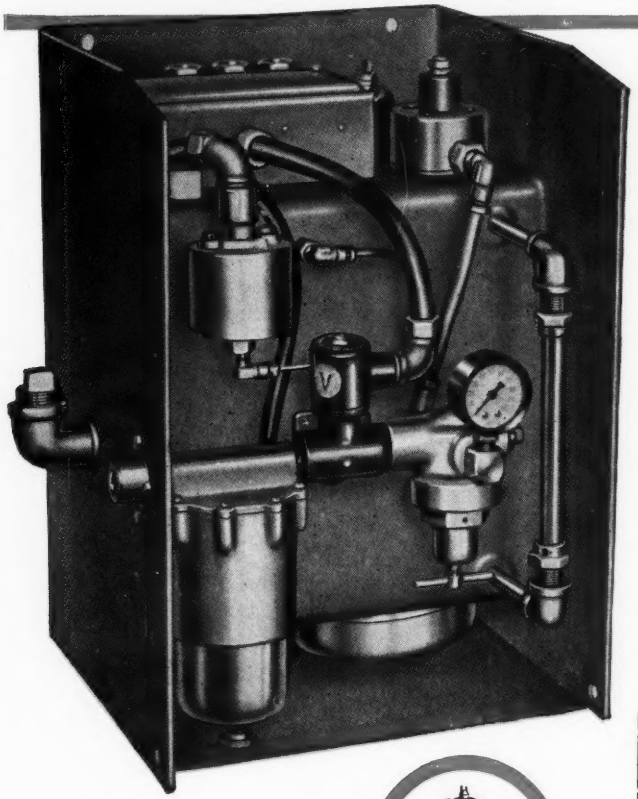

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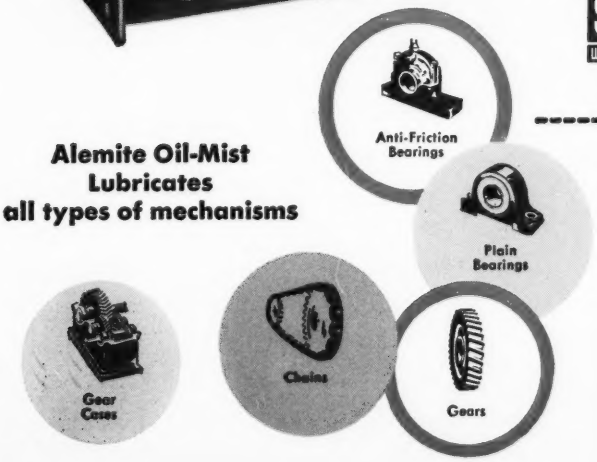

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